Applied Research Laboratory

Technical Report

A THEORETICAL ANALYSIS OF AIRBORNE SOUND TRANSFER FROM A RESILIENTLY MOUNTED MACHINE TO ITS FOUNDATION

by

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C.B. Burroughs

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NOMENCLATURE

c	speed of acoustic wave propagation
D	bending rigidity
E	modulus of elasticity
f	frequency (Hz)
F	input force
$G(r/r_s)$	Green's function
h	thickness of the plate
i	square root of -1
k	acoustic wavenumber or stiffness
k_p	free bending wavenumber in the infinite plate
L_x	length of plate A in the x direction
L_y	length of plate A in the y direction
x_o, y_o	drive point on plate A
z	distance between plates
w	plate response
γ	ratio of specific heats = 1.4 for air
η	damping factor
μ	viscosity of fluid
ν	Poisson's ratio
ρ	density
ω	frequency (rad/s)
ω_o	resonance frequency

Chapter 1

INTRODUCTION

Resilient mounts are often used to reduce the propagation of unwanted vibration from a machine to its foundation or to reduce the transmission of motion of a foundation to vibration sensitive equipment. Most research on and development of resilient mounting systems has focused on the structureborne paths through the mounts. The often neglected airborne path between the vibrating machine and foundation is considered in this thesis by developing an analytic model for the airborne propagation of vibrations from a finite elastic plate to a parallel infinite plate. In the model, the unsteady pressures generated by vibration of the upper plate are propagated to the surface of the infinite plate where the incident unsteady pressures excite the infinite plate into vibration. The airborne transmission loss is then computed as the ratio of the amplitudes of the vibrations of the two plates.

A simple diagram of a mounting system using four rubber mounts is shown in Figure 1.1. Using four-pole parameters as developed by Molloy (1), the transmissibility of vibration from a vibrating rigid body foundation with uniform velocity to an unconstrained mass is given by

$$T = \frac{v_2}{v_1} = \frac{i\omega \frac{C}{m} + \omega_o^2}{i\omega \frac{C}{m} + \omega_o^2 - \omega^2} , \qquad 1.1$$

where the undamped natural frequency is given by

$$\omega_o = \sqrt{\frac{k}{m}} \qquad 1.2$$

and

C = 4 times the damping factor for one mount

k = 4 times the stiffness of one mount

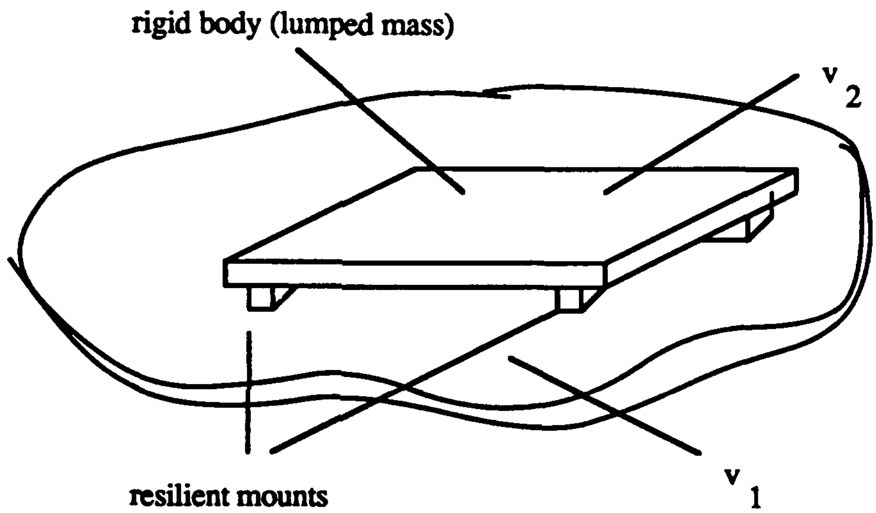

Figure 1.1 Simple diagram of a mounting system

m = the mass of the mounted item.

Using equation 1.1, the vibration transmission losses for propagation through mounts with different natural frequencies are plotted in Figure 1.2. According to equation 1.1, the transmission loss increases indefinitely as one over the frequency squared at frequencies much greater than the natural frequency. However, measurements on single stage mounting systems have rarely shown transmission losses as high as the predicted losses given in Figure 1.2.

There are several reasons why the transmission losses predicted by equation 1.1 are not achieved in reality. Some of these are:

1. Structureborne flanking transmission paths through mechanical connections to the mounted machinery, such as piping, electrical connections and duct work.
2. Wave effects in the mounts at high frequencies where resonances inside the mounts occur increase the transmissibility through the mounts. This effect has been previously studied; see, for example, Snowdon (2).
3. Airborne transmission from the vibrating structure across the mounts to the vibration sensitive structure or equipment.

Because the last effect is often neglected in mount design and because the airborne path may be a significant factor in the transmissibility for single stage mounts at high frequencies, it will be the subject of this thesis.

Heckl (3) measured the vibration transmission loss from a wall to a plate parallel to the wall with and without mounts. The transmission loss was first measured with the plate mounted on rubber mounts having a fundamental resonance of 40 Hz. The mounts were completely removed and the transmission loss measurements repeated with the same distance between the plate and the wall. Figure 1.3 shows the transmission losses

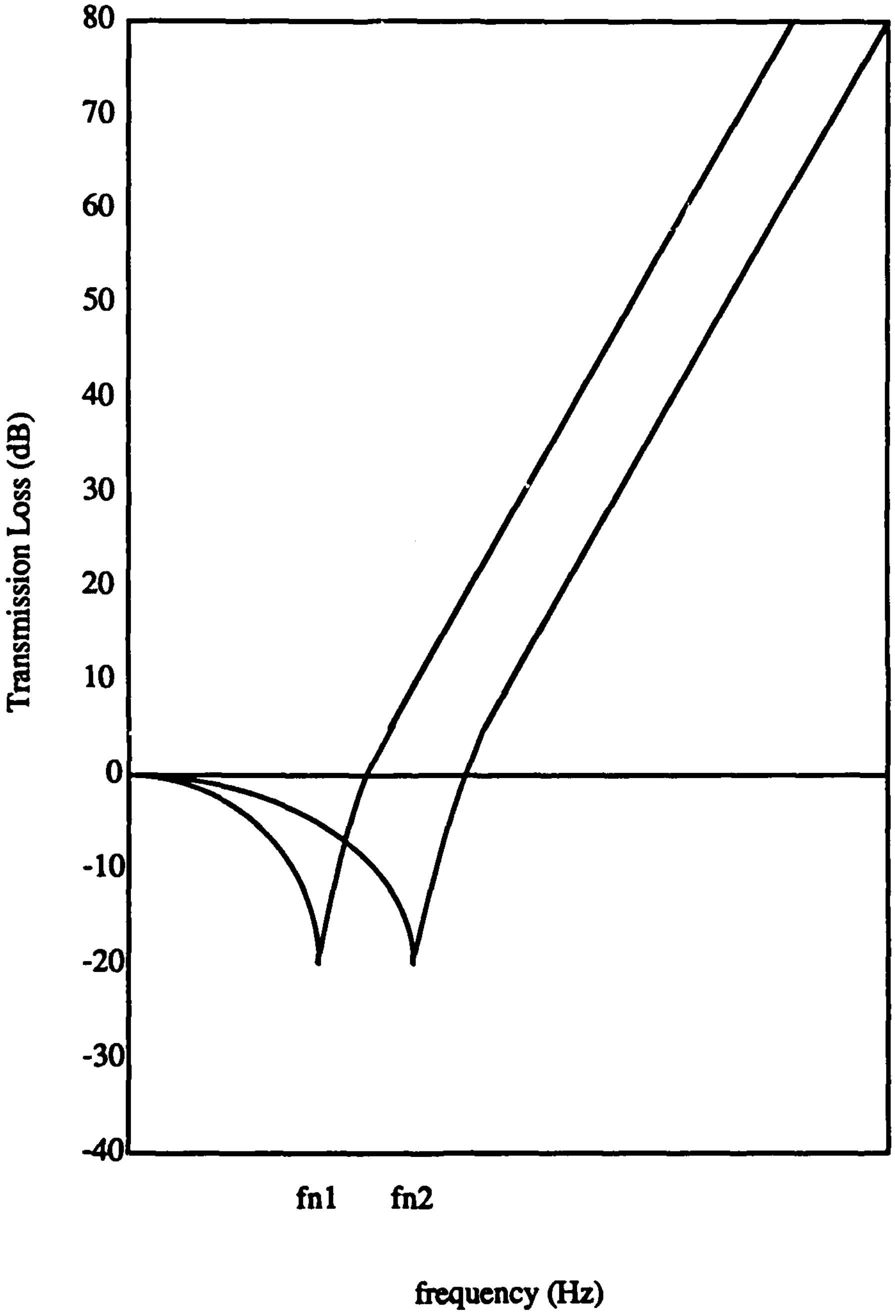

Figure 1.2 Transmission loss for propagation through resilient mounts

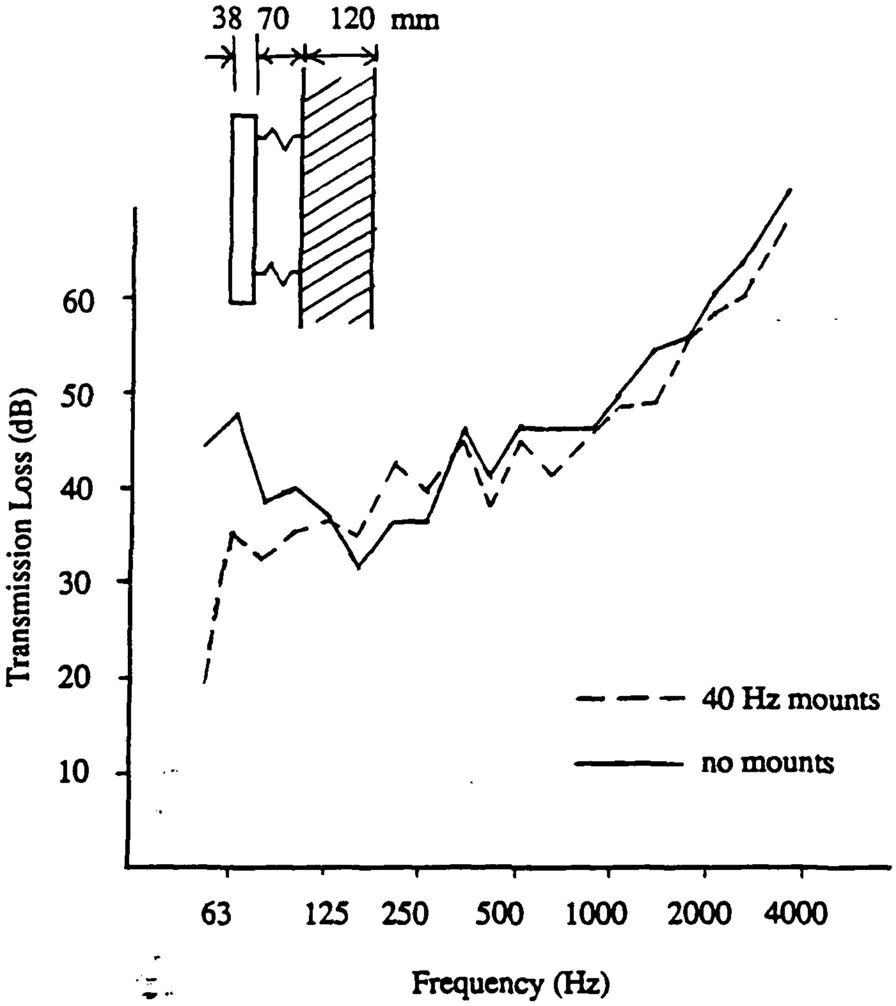

Figure 1.3 Transmission loss for a plate mounted on a wall with and without rubber mounts (3)

between the plate and the wall with and without the mounts. Above 100 Hz, the vibration transmission losses with the mounts are nearly equal to the losses without the mounts. This suggests that energy transmitted between the plate and the wall via the airborne path is nearly equal to the energy transmitted from the plate to the wall via the structureborne path through the mounts.

Ungar (4) treated the air beneath the isolated equipment as adding stiffness to the stiffness of the resilient mounts, thus increasing the stiffness of the mounting system. The resonance frequency of the entire system including the resilient mounts and the entrapped air is:

$$f_n = \left(\frac{1}{2\pi}\right)\sqrt{(k_a + k_s)/m} \qquad 1.3$$

$$= \sqrt{f_a^2 + f_s^2}$$

In the absence of air the resonance frequency f_s of the mounting system is:

$$f_s = \left(\frac{1}{2\pi}\right)\sqrt{k_s/m} \qquad 1.4$$

where

k_s = the static stiffness of the mounts

m = the mass of the mounted item.

Without the mounts, the resonance frequency f_a of the entrapped air is:

$$f_a = \left(\frac{1}{2\pi}\right)\sqrt{k_a/m} \qquad 1.5$$

$$k_a = \gamma p_o/d$$

where

k_a = the stiffness of the air

γ = ratio of specific heats = 1.4 for air

p_o = the ambient air pressure

d = the air gap thickness

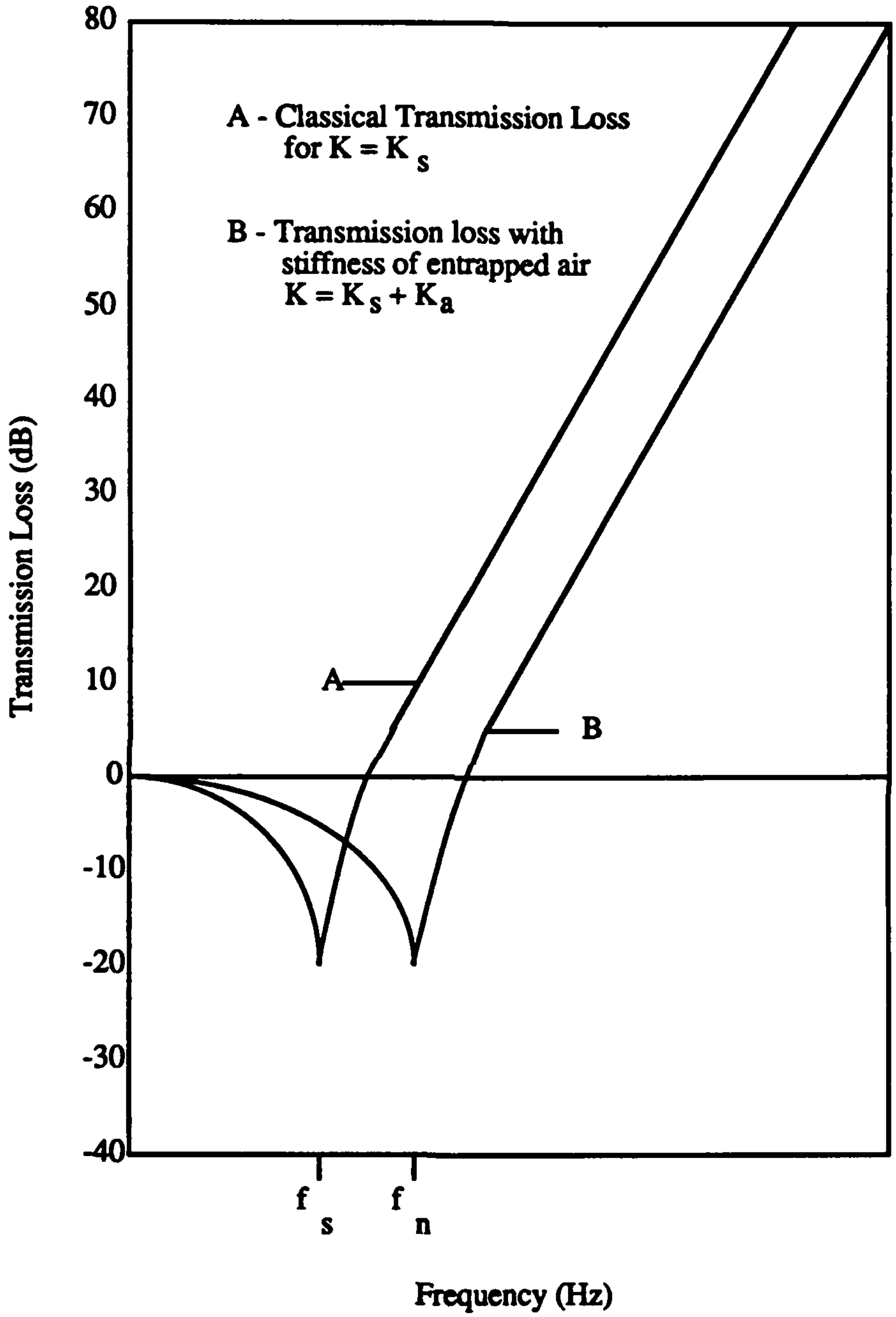

Figure 1.4 Transmission loss taking into account the stiffness of the air beneath the mounted item

Figure 1.4 shows the classical transmission loss curve without the air (curve A) and the transmission loss with the combined stiffness of the mounts and entrapped air (curve B). These curves indicate that the airborne path may be significant in the transmission across resilient mounts.

The results in Figure 1.4 are only valid when the air is entrapped. If the space between the mounted equipment and its foundation is vented, then the inertia and viscosity of the air must be taken into account when calculating the transmission loss. If the air is allowed to escape freely from under the edges of the mounted equipment, the contribution by the air to the total stiffness may be less at low frequencies.

The above analysis does not apply at higher frequencies because it assumes that the base of the isolated equipment moves as a rigid body. At higher frequencies the equipment and foundation no longer behave as rigid bodies.

To include the effects of the venting of air in between the mounted equipment and its foundation, and the elastic vibratory motion of the mounted equipment and foundation, an analytic model for the airborne transmission between parallel elastic plates is developed in this thesis. In Chapter 2, the mathematical model for the response of a finite plate to a point excitation and the airborne transmission loss from the finite vibrating plate to the vibration response of a parallel infinite elastic plate is derived. Predictions of the airborne transmission loss are presented in Chapter 3 and compared to predicted transmission losses through resilient mounts. Also the dependance of the airborne transmission losses on model parameters, such as the separation distance between the plates, thickness of the infinite plate, size of the finite plate, frequency and damping in the plates are presented. Conclusions and recommendations for additional research are given in Chapter 4.

Chapter 2

ANALYTIC MODEL

2.1. Introduction

In the analytic model of the airborne transmission from a vibrating piece of machinery to its supporting structure, derived here, the foundation (receiver structure) is modeled by an infinite flat plate (plate B) and the machinery (source structure) is modeled as a finite flat plate (plate A) parallel to the surface of the receiving structure as shown in figure 2.1. In developing this model, the following steps are taken.

1. The solution for the response of the top simply supported plate (plate A) to a point excitation is derived using classical modal expansion.
2. Euler's equation is used to relate the velocity of plate A to the unsteady pressure on the surface of plate A facing the infinite plate (plate B).
3. Using the wave equation, the pressure on the surface of plate A is propagated to the surface of plate B.
4. Fourier transforms are used to transform the pressure on the surface of plate B into wavenumber space.
5. Using the Green's function for plate B in wavenumber space, the solution for the response of plate B to the pressure on its surface is obtained as the product of the Green's function and the pressure on the surface of plate B.
6. The inverse Fourier transform is taken to obtain the spatial distribution of the response of plate B.
7. Predictions of the transmission loss are derived by taking the ratio of the averages of the vibration responses computed for plates A and B.

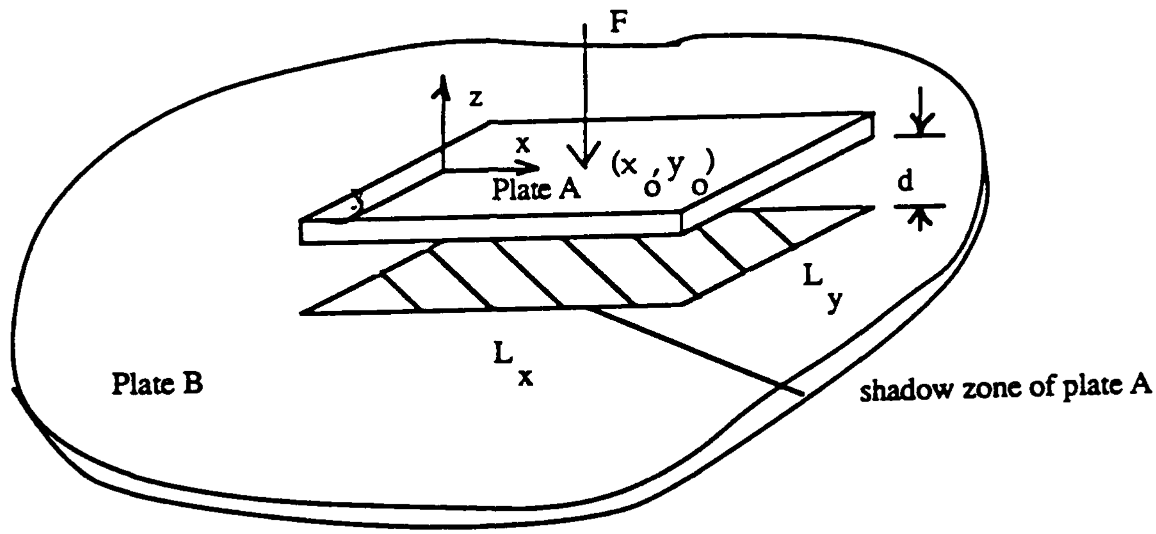

Figure 2.1 Infinite flat plate with a finite flat plate parallel to it

2.2. Assumptions

One of the assumptions in the development of the analytic model is that the impedances of the plates are much larger than the acoustic impedance of the air. With this assumption, a) the effect of air loading on the vibration of plate A will be neglected, b) the response of plate A to pressure radiated by plate B back to plate A will be neglected, and c) the pressure on the surface of plate B will be the "blocked" pressure, i.e. the motion of plate B will have little impact on the pressure on the surface of plate B. Therefore, with the impedances of the plates much larger than the acoustic impedance, the solution for plate vibrations decouple from the air pressures. Also standing waves in between the two plates will not be included in the model.

2.3. Mathematics

2.3.1 Vibration Response of Plate A

For Plate A, simply supported, and driven by a point source located at x_o and y_o as shown in figure 2.1, the plate equation is

$$D_a \nabla^4 w_a - \rho_a h_a \omega^2 w_a = F\delta(x-x_o)\delta(y-y_o) \quad , \qquad 2.1$$

where the bending rigidity of plate A is given by

$$D_a = \frac{E_a h_a{}^3}{12(1-\nu^2)} \qquad 2.2$$

where ρ_a is the density of plate A,

h_a is the thickness of plate A,

E_a is the modulus of elasticity of plate A, and

ν is Poissons ratio.

For a simply supported plate the boundary conditions are:

$$\left.\begin{aligned} w_a &= 0 \\ \frac{\partial^2 w_a}{\partial x^2} &= 0 \end{aligned}\right\} \text{ for } x = \pm L_x/2$$

and

$$\left.\begin{aligned} w_a &= 0 \\ \frac{\partial^2 w_a}{\partial y^2} &= 0 \end{aligned}\right\} \text{ for } y = \pm L_y/2 \tag{2.3}$$

With opposing edges simply supported, the solution is separable, so that

$$w_a(x,y) = X(x)Y(y) \quad . \tag{2.4}$$

To satisfy the boundary conditions given by equations 2.3,

$$X(x) = \cos\left[\frac{(2m+1)\,\pi\, x}{L_x}\right]$$

$$Y(y) = \cos\left[\frac{(2n+1)\,\pi\, y}{L_y}\right] \quad . \tag{2.5}$$

The solution can then be expressed as

$$w_a(x,y) = \sum_{m=0}^{\infty} \sum_{n=0}^{\infty} A_{mn} \cos\left[\frac{(2m+1)\,\pi\, x}{L_x}\right] \cos\left[\frac{(2n+1)\,\pi\, y}{L_y}\right] \quad . \tag{2.6}$$

The resonant frequencies ω_{mn} are obtained by using equation 2.6 in the homogeneous equation 2.1,

$$\omega_{mn}^{2} = \frac{D_a}{\rho_a h_a} \left\{ \left[\frac{(2m+1)\pi}{L_x}\right]^2 + \left[\frac{(2n+1)\pi}{L_y}\right]^2 \right\}^2 . \qquad 2.7$$

Now using equation 2.6 in equation 2.1, and operating with

$$\int_{-Lx/2}^{Lx/2} \int_{-Ly/2}^{Ly/2} \cos\left[\frac{(2m'+1)\,\pi\, x}{L_x}\right] \cos\left[\frac{(2n'+1)\,\pi\, y}{L_y}\right] dx\, dy$$

and using orthogonality yields

$$A_{mn} = \frac{4\,F \cos\left[\frac{(2m+1)\pi x_o}{L_x}\right] \cos\left[\frac{(2n+1)\pi y_o}{L_y}\right]}{M_p\,[\,\omega_{mn}^{2} - \omega^2\,]} \qquad 2.8$$

where $M_p = \rho_a h_a\, L_x\, L_y$.

Using equation 2.8 in equation 2.6,

$$w_a(x,y) =$$

$$\frac{4F}{M_p}\sum_{m=0}^{\infty}\sum_{n=0}^{\infty} \frac{\cos\left[\frac{(2m+1)\pi x_o}{L_x}\right]\cos\left[\frac{(2n+1)\pi y_o}{L_y}\right]\cos\left[\frac{(2m+1)\pi x}{L_x}\right]\cos\left[\frac{(2n+1)\pi y}{L_y}\right]}{[\,\omega_{mn}^{2} - \omega^2\,]} .$$

2.9

Taking the Fourier transform of equation 2.9

$$w_a(k_x,k_y) = \frac{4F}{M_p}\sum_{m=0}^{\infty}\sum_{n=0}^{\infty} \frac{\cos\left[\frac{(2m+1)\pi x_o}{L_x}\right]\cos\left[\frac{(2n+1)\pi y_o}{L_y}\right]}{[\,\omega_{mn}^{2} - \omega^2\,]}\, I_{mn}(k_x,k_y) \qquad 2.10$$

where

$$I_{mn}(k_x,k_y) = \int_{-\infty}^{\infty}\int_{-\infty}^{\infty} \cos\left[\frac{(2m+1)\pi x}{L_x}\right]\cos\left[\frac{(2n+1)\pi y}{L_y}\right] e^{-ik_x x}\, e^{-ik_y y}\, dx\, dy \quad . \qquad 2.11$$

$I_{mn}(k_x,k_y)$ is evaluated in Appendix A.

2.3.2. Acoustic Field Between Plates

The unsteady pressure in the air between plate A and plate B satisfies the wave equation which, for harmonic time dependence $e^{-i\omega t}$, becomes the Helmholtz equation,

$$\nabla^2 p + k^2 p = 0 \qquad 2.12$$

where $k = \frac{\omega}{c}$ is the acoustic wavenumber.

Taking the Fourier transform of equation 2.12 yields

$$(k^2 - k_x{}^2 - k_y{}^2 + \frac{\partial^2}{\partial z^2})\, p\, (k_x,k_y;z) = 0 \quad . \qquad 2.13$$

The solution to equation 2.13 is

$$p\ (k_x,k_y;z) = A\ e^{i(k^2-k_x{}^2-k_y{}^2)^{1/2} z} \quad . \qquad 2.14$$

At the surface of plate A, the velocities of the air and the plate are equal, so that, from Euler's equation with harmonic time dependence, we have

$$\rho_o \omega^2 w_a = \frac{\partial p}{\partial z}\ \Big]_{z=0} \quad . \qquad 2.15$$

Taking the Fourier transform yields

$$\rho_o \omega^2 w_a(k_x,k_y) = \frac{\partial}{\partial x} p(k_x,k_y;z) \Big]_{z=0} \quad . \qquad 2.16$$

Using equation 2.14,

$$A = \frac{-i\rho_o \omega^2 w_a(k_x,k_y)}{(k^2-k_x^2-k_y^2)^{1/2}} \quad . \qquad 2.17$$

So that equation 2.14 for the pressure radiated by plate A in wavenumber space becomes

$$p\,(k_x,k_y;z) = \frac{-i\rho_o \omega^2 w_a(k_x,k_y)}{(k^2-k_x^2-k_y^2)^{1/2}}\, e^{i(k^2-k_x^2-k_y^2)^{1/2} z} \quad . \qquad 2.18$$

Using equation 2.10 in equation 2.18,

$$p\,(k_x,k_y;z) = \frac{-i4F\rho_o \omega^2}{M_p(k^2-k_x^2-k_y^2)^{1/2}}\, e^{i(k^2-k_x^2-k_y^2)^{1/2} z} \sum_{m=0}^{\infty}\sum_{n=0}^{\infty} \frac{\cos\left[\frac{(2m+1)\pi x_o}{L_x}\right]\cos\left[\frac{(2n+1)\pi y_o}{L_y}\right]}{[\,\omega_{mn}^2 - \omega^2\,]}\, I_{mn}(k_x,k_y) \quad . \qquad 2.19$$

This is the pressure radiated by plate A into a free space, as a function of wavenumber, k_x and k_y, at a distance z from the plate. The pressure driving plate B will be the "blocked" pressure which is twice the free-field pressure given in equation 2.19.

2.3.3. Vibration Response of Plate B

The response of plate B to the pressure on its surface is given by

$$w_b(x',y') = 2 \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} p(x,y;d)\, G(x',y'/x,y)\, dx\, dy \quad , \tag{2.20}$$

where $w_b(x',y')$ is the velocity of plate B, p(x,y;d) is the free field pressure at the surface of plate B, d is the separation distance between the plates, and G(x',y'/x,y) is the Green's function which satisfies

$$D_b \nabla^4 G - \rho_b h_b \omega^2 G = \delta(x-x')\, \delta(y-y') \quad . \tag{2.21}$$

Equation 2.21 is the classical plate equation with harmonic time dependence and pressure of unit amplitude acting on the plate at the point x',y'. Therefore, for the infinite plate, plate B, the Green's function, G(x',y'/x,y) is the velocity of the plate at x', y' due to a unit force applied at a point x,y. Since the plate is infinite, the Green's function depends only on the separation between the point source and the receiver, and not on each location. Equation 2.20 can be written as:

$$w_b(x',y') = 2 \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} p(x,y;z)\, G(x-x',y-y')\, dx\, dy \quad . \tag{2.22}$$

Equation 2.22 is the convolution integral and its Fourier transform is

$$\mathit{w}_b(k_x,k_y) = 2\, \mathit{p}(k_x,k_y;z)\, \mathit{G}(k_x,k_y) \quad . \tag{2.23}$$

For an infinite plate, it can be assumed in equation 2.21 that x' = y' = 0, so that

$$D_b\left(\frac{\partial^4}{\partial x^4} + 2\frac{\partial^4}{\partial x^2 \partial y^2} + \frac{\partial^4}{\partial y^4}\right)G - \rho_b h_b \omega^2 G = \delta(x)\,\delta(y) \quad . \qquad 2.24$$

Taking Fourier transforms on x and y, and solving for the Green's function yields

$$G(k_x,k_y) = \frac{1}{D_b\left[(k_x{}^2 + k_y{}^2)^2 - k_p{}^4\right]} \qquad 2.25$$

where $k_p{}^4$,the free structural wavenumber, is given by

$$k_p{}^4 = \frac{\rho_b h_b}{D_b}\omega^2 \quad . \qquad 2.26$$

Using equations 2.19 and 2.25 in equation 2.23, yields

$$w_b(k_x,k_y) = \frac{-i8F\rho_o\omega^2}{M_p} \frac{e^{i(k^2-k_x{}^2-k_y{}^2)^{1/2}z}}{D_b\left[(k_x{}^2+k_y{}^2)^2-k_p{}^4\right](k^2-k_x{}^2-k_y{}^2)^{1/2}} \sum_{m=0}^{\infty}\sum_{n=0}^{\infty} \frac{\cos\left[\frac{(2m+1)\pi x_o}{L_x}\right]\cos\left[\frac{(2n+1)\pi y_o}{L_y}\right]}{[\,\omega_{mn}{}^2 - \omega^2\,]} I_{mn}(k_x,k_y) \quad . \qquad 2.27$$

Taking the inverse Fourier transform, yields the final solution for the response of plate B,

$$w_b(x,y) = \frac{-i8F\rho_o\omega^2}{(2\pi)^2 M_p D_b} \sum_{m=0}^{\infty}\sum_{n=0}^{\infty} \frac{\cos\left[\frac{(2m+1)\pi x_o}{L_x}\right]\cos\left[\frac{(2n+1)\pi y_o}{L_y}\right]}{[\,\omega_{mn}{}^2 - \omega^2\,]} \int_{-\infty}^{\infty}\int_{-\infty}^{\infty} \frac{e^{i(k^2-k_x{}^2-k_y{}^2)^{1/2}z}\, I_{mn}(k_x,k_y)}{\left[(k_x{}^2+k_y{}^2)^2-k_p{}^4\right](k^2-k_x{}^2-k_y{}^2)^{1/2}}\, e^{ik_x x}\, e^{ik_y y}\, dk_x\, dk_y \quad . \qquad 2.28$$

Equation 2.28 has three singularities; one at the resonance frequencies of plate A, $\omega_{mn} = \omega$, one at the acoustic wavenumber in the air between the plates, $k^2 = k_x^2 - k_y^2$, and one at the free bending wavenumber in plate B, $k_p^4 = (k_x^2 + k_y^2)^2$. In order to solve this equation these singularities must be addressed.

At resonance frequency, the response of plate A becomes infinite because there is no damping in the model. In reality, there is always some damping, which is added to the model by making the bending rigidity of plate A complex,

$$D_a^* = D_a (1 + i\eta_a) \quad , \qquad 2.29$$

where η_a is the damping loss factor of plate A.

The singularity at the acoustic wavenumber results from taking a Fourier transform over the infinite spatial domain of plate B. With no dissipation, the acoustic pressure waves in the air will propagate parallel to plate B out to infinity, so that energy will sum to infinity when a sum is taken over all space in the infinite Fourier transform. In reality, there will be some dissipation of the acoustic energy propagating parallel to plate B. Also, energy that propagates parallel to plate B should couple less to the response of plate B then energy that propagates directly across the space between the plates and is incident on plate B at non-grazing angles. Therefore dissipation can be added to the acoustic propagation to make the model more realistic and remove the singularity at the acoustic wavenumber, without impacting the results at non-acoustic wavenumbers.

Pierce (5 eqn. 10-8.9b) gives the acoustic wavenumber with absorption as:

$$k^* = \frac{\omega}{c} + i\alpha_{cl}' \quad , \qquad 2.30$$

where the classical absorption constant is given by (from Pierce (5 eqn.10-2.12))

$$\alpha_{cl}' = \frac{\omega^2}{c^3} \frac{\mu}{2\rho_o} \left(\frac{4}{3} + \frac{\mu_B}{\mu} + \frac{\gamma - 1}{Pr} \right) \tag{2.31}$$

where μ is the viscosity of air,

ρ_o is the density of air,

μ_B is the bulk viscosity of air,

γ is the ratio of specific heats = 1.4 for air, and

Pr is the Prandtl number for air.

Ignoring effects of bulk viscosity (μ_B) and using equation 2.31 in equation 2.30,

$$k^* = k \left\{ 1 + i \left[\frac{\omega}{c^2} \frac{\mu}{2\rho_o} \left(\frac{4}{3} + \frac{\gamma - 1}{Pr} \right) \right] \right\} \quad . \tag{2.32}$$

Equation 2.32 was used for the acoustic wavenumber in equation 2.28.

The singularity at the free bending wavenumber in plate B can be removed by adding damping to the plate in a manner similar to the damping added to plate A. This damping has the same effects as the dissipation added to the air between the plates. With damping, the bending wavenumber can be written as

$$(k_p^*)^4 = k_p^4 (1 + i\eta_b) \quad , \tag{2.33}$$

where η_b is the damping loss factor of plate B.

2.3.4. The Discrete Inverse Fourier Transform

To evaluate the integral in equation 2.28 using numerical methods, a two dimensional discrete inverse Fourier transform was developed based on the continuous infinite inverse Fourier transform as defined by Brigham (6),

$$f(x,y) = \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} F(k_x,k_y)\, e^{ik_x x}\, e^{ik_y y}\, dk_x\, dk_y \quad . \qquad 2.34$$

This integral is approximated by limiting the range of integration, so that

$$f(x,y) \approx \int_{-K_x/2}^{K_x/2} \int_{-K_y/2}^{K_y/2} F(k_x,k_y)\, e^{ik_x x}\, e^{ik_y y}\, dk_x\, dk_y \quad , \qquad 2.35$$

where K_x and K_y are at least twice the value of k_p as calculated from equation 2.26 or twice the acoustic wavenumber k which ever value is larger at the frequency of interest. These limits were selected so that the range of integration includes all of the points where singularities would exist in the absence of added damping and dissipation and so that the exponential term $e^{i(k^2-k_x^2-k_y^2)^{1/2}}$ decays to a small number at the limits of integration. Equation 2.35 can be written

$$f(x,y) = e^{-ixK_x/2}\, e^{-iyK_y/2} \int_0^{K_x} \int_0^{K_y} F(k_x - K_x/2, k_y - K_y/2)\, e^{ik_x x}\, e^{ik_y y}\, dk_x\, dk_y \, . \qquad 2.36$$

This integral can be evaluated using the rectangular rule with spacing $h_{kx} = \frac{K_x 2\pi}{N}$ and $h_{ky} = \frac{K_y 2\pi}{N}$ so

$$f(x,y) \approx \frac{1}{(2\pi)^2} e^{-ixK_x/2} e^{-iyK_y/2} h_{kx} h_{ky}$$

$$\sum_{m=0}^{N-1} \sum_{n=0}^{N-1} e^{ixmh_{kx}} e^{iynh_{ky}} F(mh_{kx} - K_x/2, nh_{ky} - K_y/2) \quad . \qquad 2.37$$

By setting $x = \frac{2\pi j_x}{K_x}$ and $y = \frac{2\pi j_y}{K_y}$ where $j_x = 0,1,...N-1$ and $j_y = 0,1,...N-1$. Equation 2.37 becomes

$$f(\frac{2\pi j_x}{K_x}, \frac{2\pi j_y}{K_y}) = \frac{1}{(2\pi)^2} (-1)^{j_x} (-1)^{j_y} h_{kx} h_{ky}$$

$$\sum_{m=0}^{N-1} \sum_{n=0}^{N-1} e^{\frac{i(2\pi)^2 j_x m}{N}} e^{\frac{i(2\pi)^2 j_y n}{N}} F(mh_{kx} - K_x/2, nh_{ky} - K_y/2) \quad .$$

2.38

Thus, after scaling by the factors $\frac{1}{(2\pi)^2} (-1)^{j_x} (-1)^{j_y} h_{kx} h_{ky}$, the discrete inverse Fourier transform given by equation 2.38 approximates the continuous inverse Fourier transform given by equation 2.35.

To calculate the plate response, the infinite summation in equation 2.28 can be solved by starting the summation at the resonance frequencies, ω_{mn} which are nearest the frequency of interest. With this approach, the summation converges more rapidly than beginning with the lowest mode numbers. This is seen by observing equation 2.28. When a resonance frequency is chosen which is not near the resonance frequency of interest, the $(\omega_{mn}{}^2 - \omega^2)$ term in the denominator becomes large and the contribution to the sum becomes small. When solving equation 2.28 an iterative process was used. The

summation was taken using the resonance frequencies near the frequency of interest until the contribution to the summation became negligible. The same process can be used to solve the infinite summation in equation 2.9.

2.3.5. Transmission Loss between Plate A and Plate B

Transmission loss is given by

$$TL = 10 \log_{10} \frac{(A_A)^2}{(A_B)^2} \quad , \qquad 2.39$$

where A_A is the average of the magnitude of the velocity of plate A vibration, computed from equation 2.9 and A_B is the average of the magnitude of the velocity of plate B vibration, computed from equation 2.28.

For purposes of computation, the transmission loss for the vibration response of plate B is computed in the shadow zone of plate A as illustrated in Figure 2.1. By computing the vibration amplitude at N points on plate A and M points on plate B, the transmission loss is determined using the following equation:

$$TL = 10 \log_{10} \frac{\frac{1}{N} \sum_{i=1}^{N} (A_A{}^i)^2}{\frac{1}{M} \sum_{j=1}^{M} (A_B{}^j)^2} \quad . \qquad 2.40$$

N and M are chosen so that there are three points per wavelength at the highest frequency.

Chapter 3

RESULTS

The equations developed in the previous chapter are used to predict airborne transmission losses for plate configurations which are representative of typical machine/ foundation arrangements. Table 3.1 shows the input values used in the equations. For various input parameters, surface plots of the displacements of plates A and plate B were obtained to show the motion of the plates and graphs of airborne transmission loss between plates A and B as a function of frequency are presented.

3.1. Surface Plots

Figures 3.1 through 3.6 are surface plots of the motion of plates A and B for various frequencies and damping. These surface plots are essentially a snap shot of the displacement of the plate. The surface plots for plate A were obtained using equation 2.9 in a FORTRAN computer program. The surface plots for plate B were obtained using equation 2.28 in a FORTRAN computer program. The discrete inverse Fourier transform as described in equation 2.38 was used to solve the integral part of equation 2.28.

In Figures 3.1 through 3.6 the figure with the "a" suffix is the surface plot of plate A and the figure with the "b"suffix is the surface plot of plate B. The x and y ordinates are the dimensions of the plate in meters (m). The vertical axis is the amplitude of displacement of the plate in meters. The parameters used to make the predictions presented in Figures 3.1 through 3.6 are listed in Table 3.1 and shown on the surface

Table 3.1
Input parameters for calculation of surface plots and transmission loss curves

Parameters of Finite Plate (Plate A)

dimensions of plate..........L_x = 1.0 m L_y = 1.0 m

driving point..................x_o = 0.0 m y_o = 0.0 m

driving force....................F = 1.0 N

plate thickness.................h_a = 0.001587 m

plate density....................ρ_a = 7860.0 kg/m^2

modulus of elasticity..........E_a = 200 x 10^9 N/m^2

bending rigidity..............D_a = 72.73 N/m

damping coefficient..........η_a = 0.001

Parameters of Infinite Plate (Plate B)

plate thickness.................h_b = *0.001587 m*

plate density....................ρ_b = 7860.0 kg/m^2

modulus of elasticity..........E_b = 200 x 10^9 N/m^2

bending rigidity..............D_b = 72.73 N/m

damping coefficient..........η_b = 0.001

Parameters for Fluid (Air)

density..........................ρ_b = 1.204 kg/m^2

viscosity.........................μ = 0.0000184 kg/ms

speed of sound.................c = 343.0 m/s

Prandtl number................Pr = 0.706

specific heat ratio.............γ = 1.4

distance between plates........z = 0.1 m

plots. Surface plots were generated for a low resonant frequency, a low non-resonant frequency and a high resonant frequency, for both high and low damping. Plate A is a square plate with a sinusoidal force of 1 Newton (N) (0.2248 lb.) applied in the center. Both plates are steel. The edges of plate A are at zero amplitude as required by the simple supports at the edges.

Figures 3.1, 3.2, 3.3, and 3.4 show plate A and plate B excited at 37.93 Hz which from equation 2.7 is the n = 0, m = 1 resonant mode of plate A. In figures 3.1 and 3.2 both plates are highly damped ($\eta_a = \eta_b = 0.1$) and in Figure 3.3 and 3.4 both plates are lightly damped ($\eta_a = \eta_b = 0.001$). Comparing Figures 3.1 and 3.3 it can be seen that the amplitude of the undamped plate A is much larger than the highly damped plate A. This illustrates the well known effect of damping at reducing the resonant response of structures. Comparing the responses of the damped and undamped plate A (Figures 3.1 and 3.3), it can be seen that the mode shapes are the same even though the maximum amplitude of the plate vibration is different.

The symmetry of the waves excited in plate B and propagating outward from the shadow zone of plate A can be seen clearly in Figure 3.2. In this figure, the exponential decay due to damping can also be seen. The amplitude of the waves in the middle of plate B directly beneath the top plate is large and the waves decay exponentially as they spread towards the edge of the plate. In Figure 3.4 for the lightly damped plate B it can be seen that the plate is fully excited. Also, because of the decrease in the amplitude of vibration of plate A, the vibration of plate B in the shadow zone is lower with high damping than with low damping.

Figures 3.5, 3.6, 3.7, and 3.8 show plate A and plate B excited at 110 Hz, which is not a resonance frequency of plate A. It is between the m = 2, n = 0 resonance at 98 Hz and the m = 2, n = 1 resonance at 129 Hz. The mode shape of the damped and undamped plate A (Figures 3.5 and 3.7) are similar to each other. The

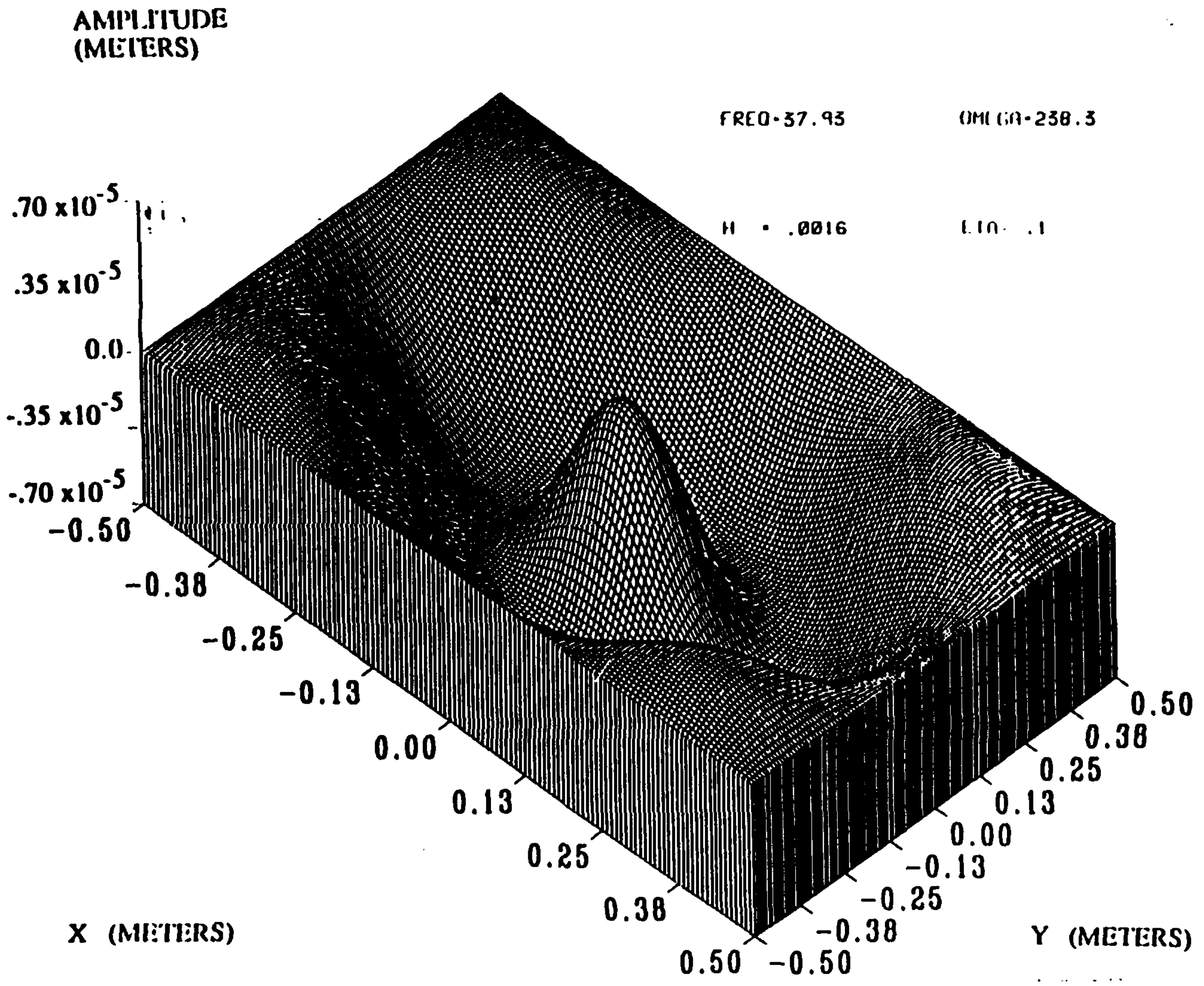

Figure 3.1 Plate A, low resonance,high damping (f=37.9Hz and $\eta = 0.1$)

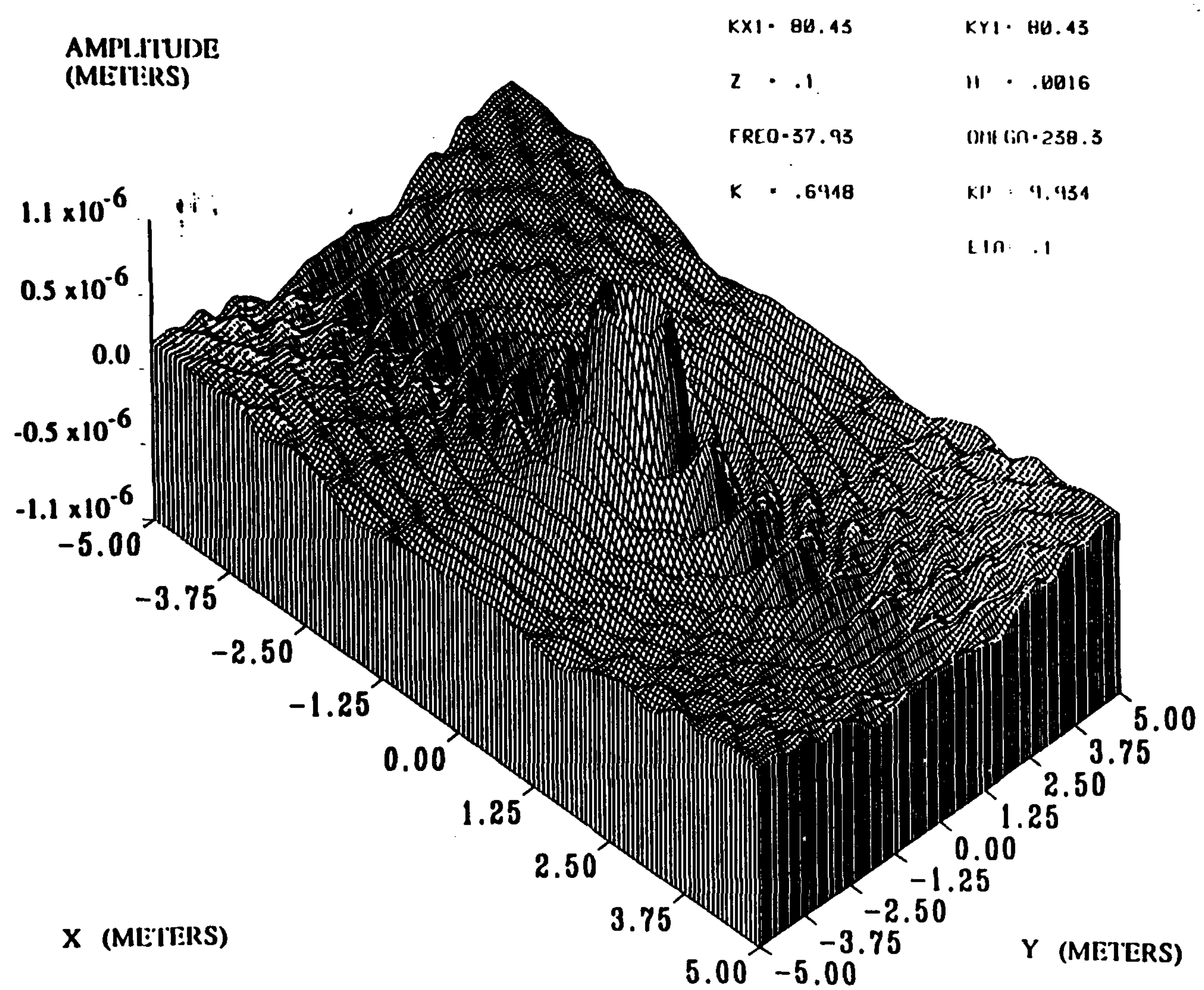

Figure 3.2 Plate B, low resonance,high damping (f=37.9Hz and $\eta = 0.1$)

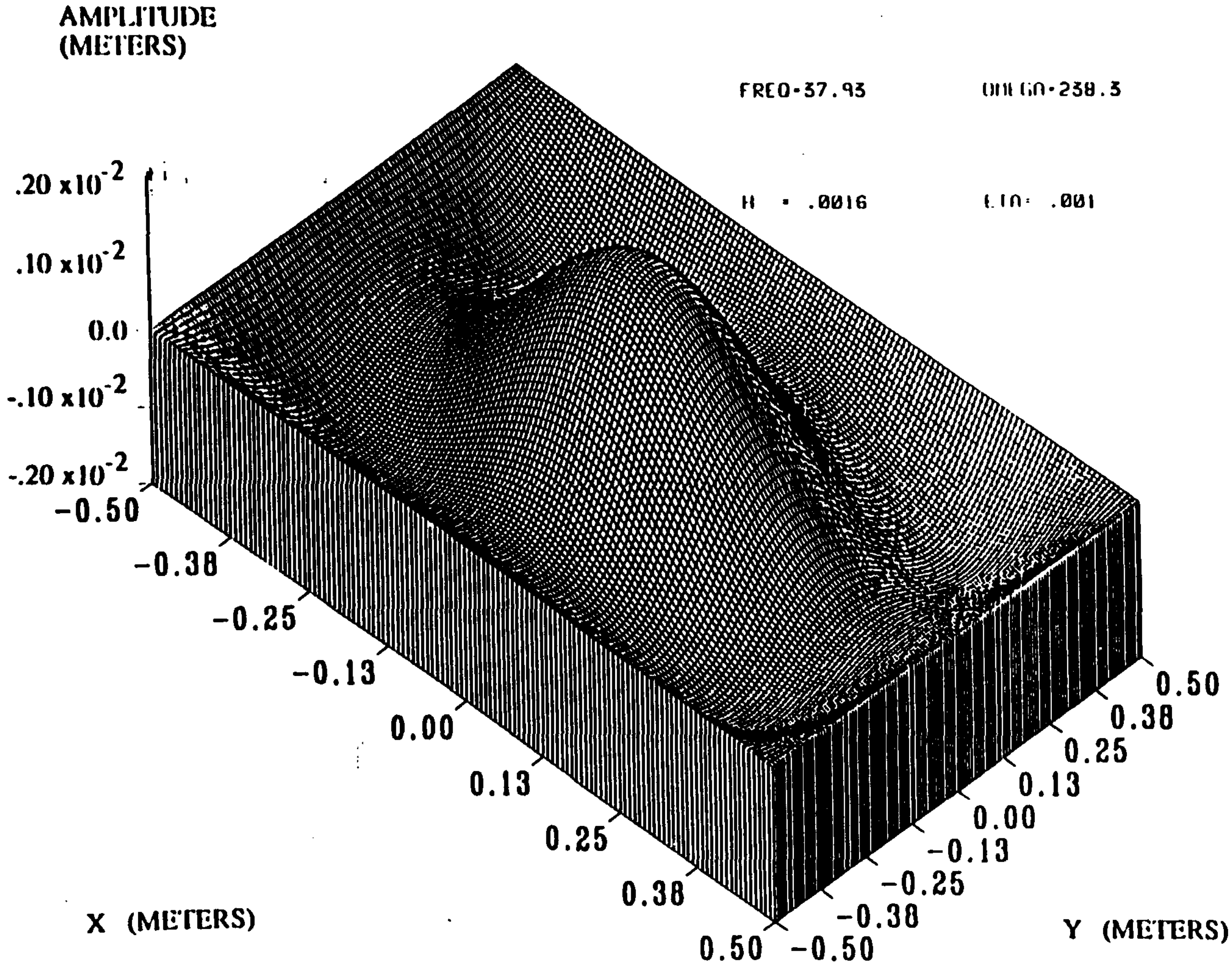

Figure 3.3 Plate A, low resonance, low damping (f=37.9 Hz and η = 0.001)

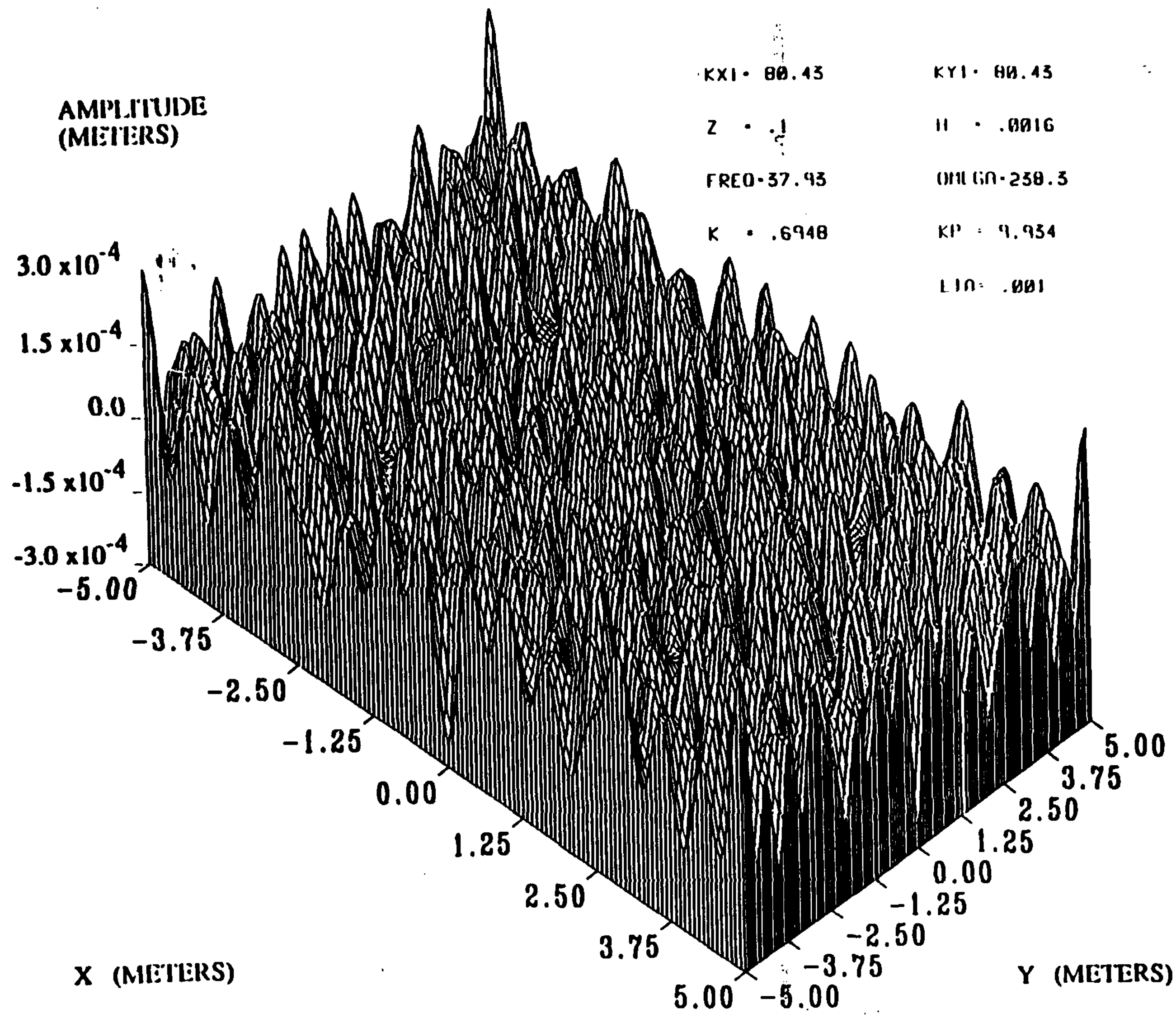

Figure 3.4 Plate B, low resonance, low damping (f=37.9 Hz and η = 0.001)

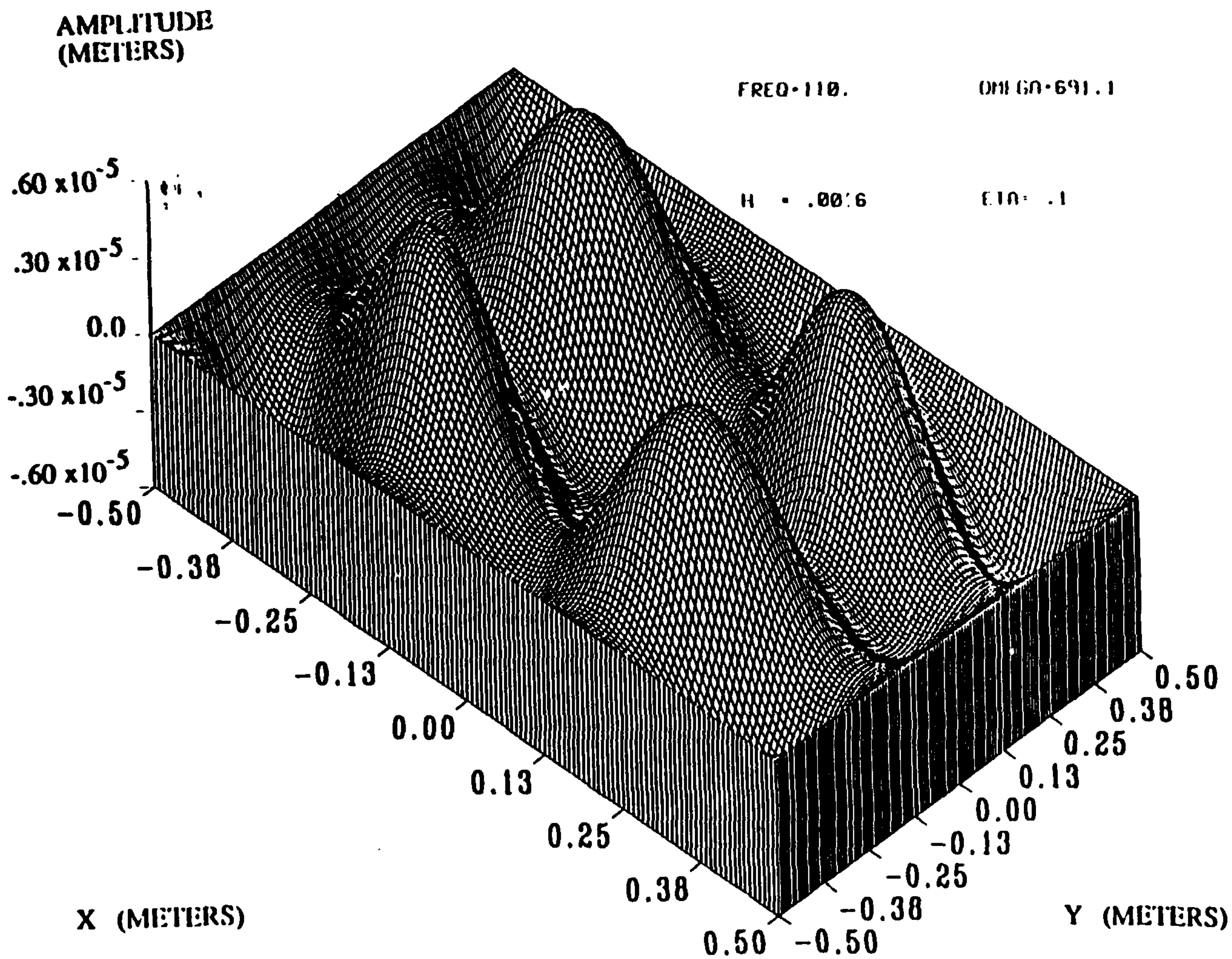

Figure 3.5 Plate A, low non-resonance, high damping (f=110.0 Hz and $\eta = 0.1$)

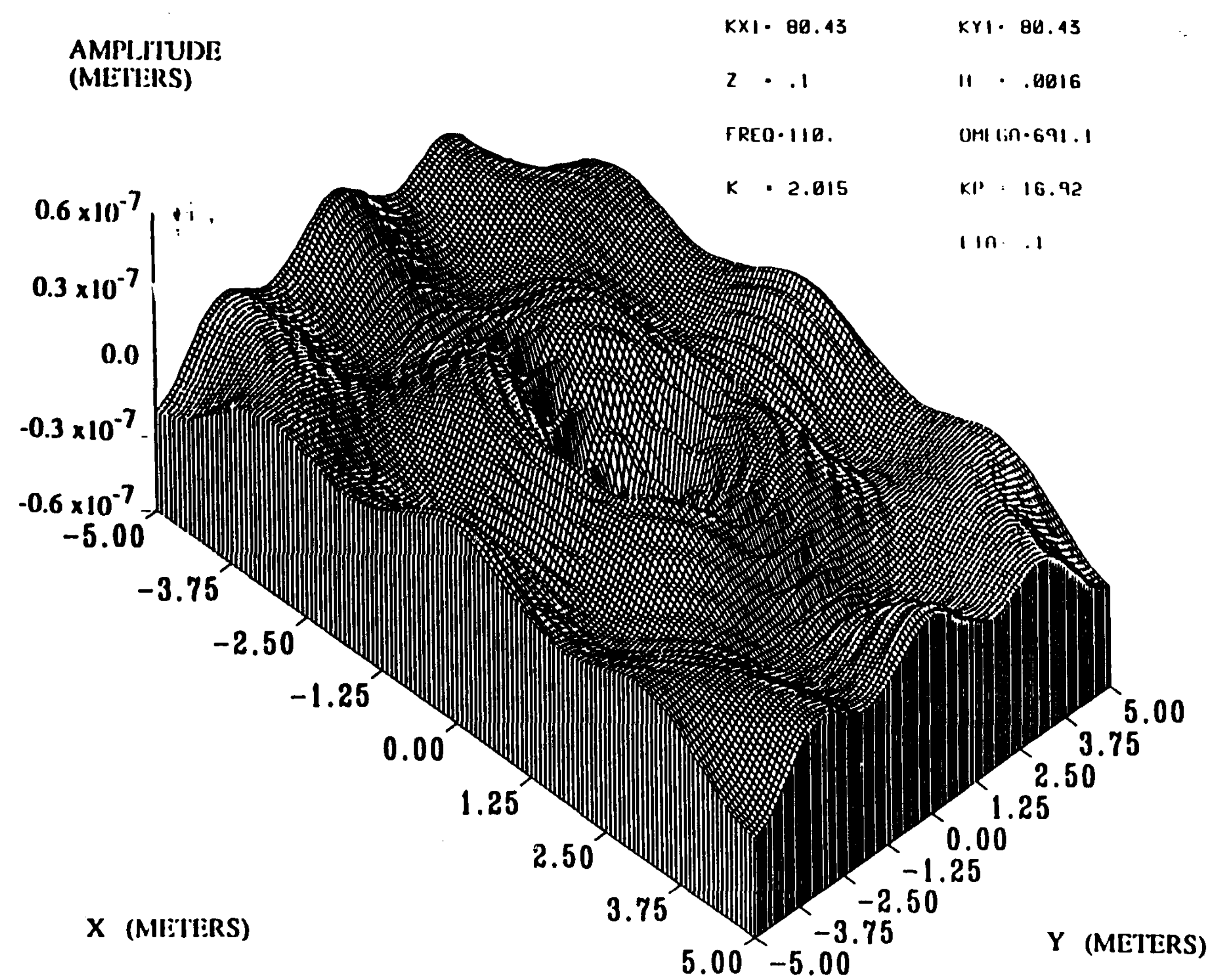

Figure 3.6 Plate B, low non-resonance, high damping (f=110.0 Hz and η = 0.1)

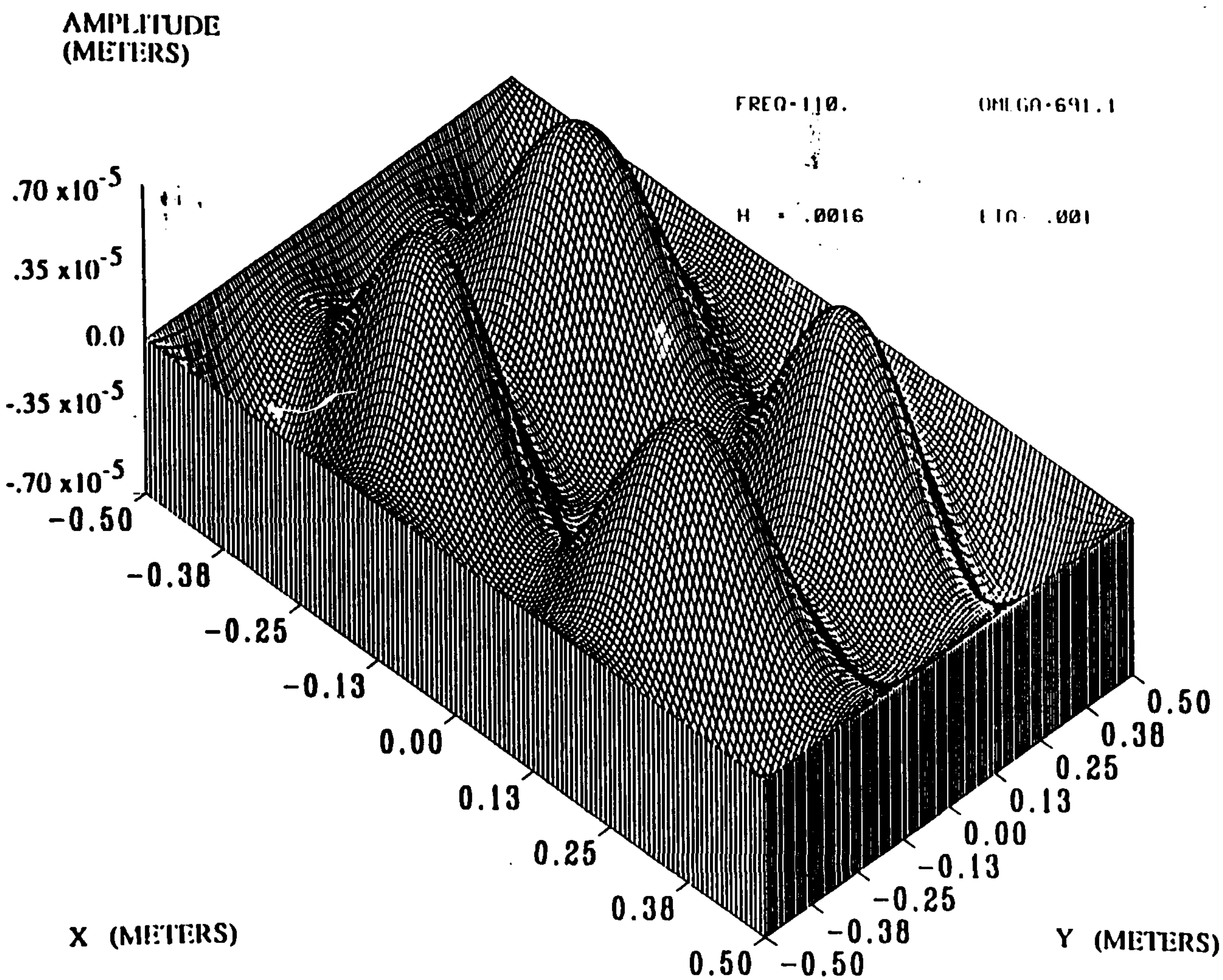

Figure 3.7 Plate A, low non-resonance, low damping (f=110.0 Hz and $\eta = 0.001$)

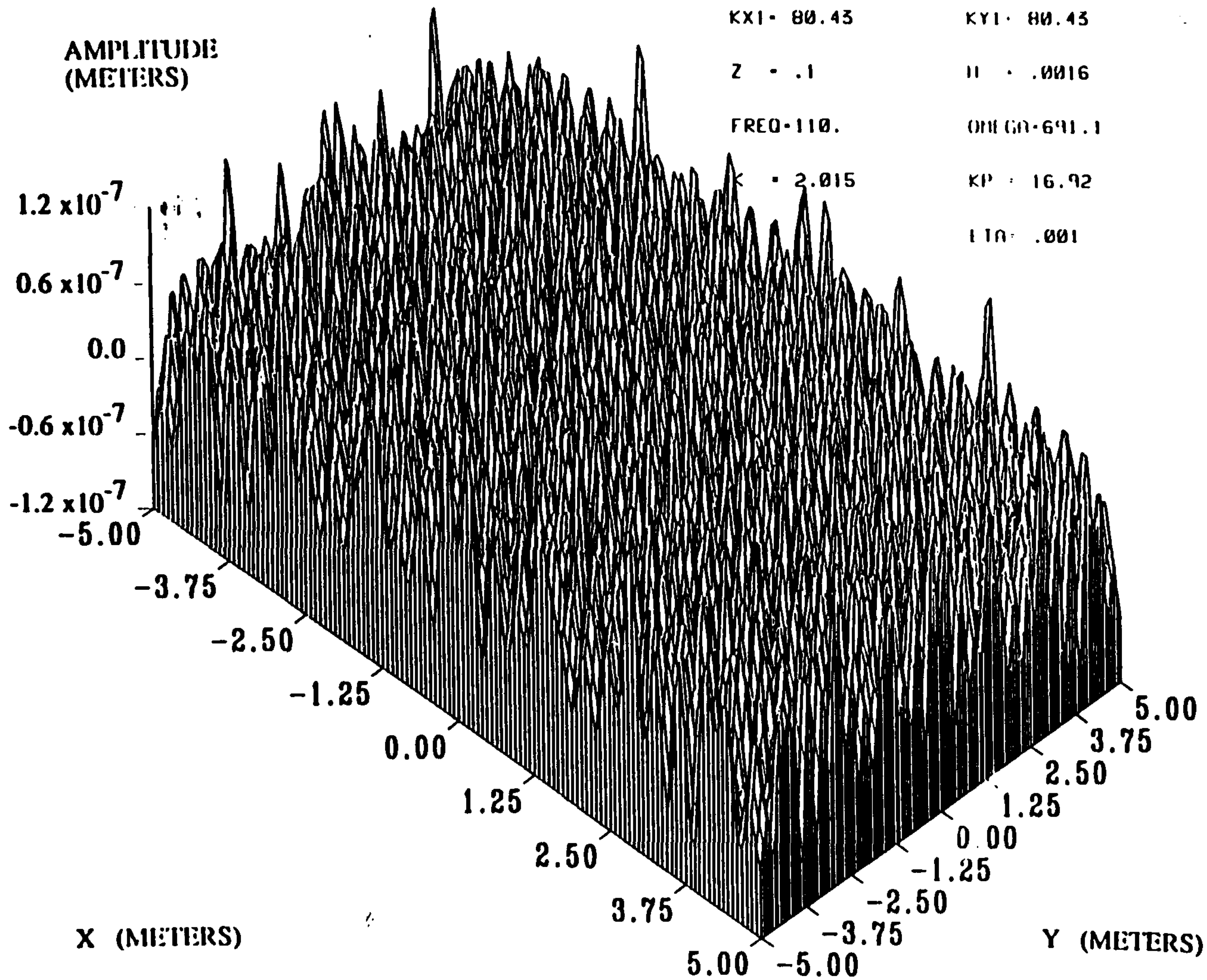

Figure 3.8 Plate B, low non-resonance, low damping (f=110.0 Hz and $\eta = 0.001$)

difference between the amplitudes of the damped and undamped plate A is not as great as at the resonance frequency (Figures 3.1 and 3.3). This is because damping in the plate is less effective at reducing structural response of a structure off resonance than at resonance.

The radial symmetry of the waves excited in plate B can again be seen in Figure 3.6 with the high damping. The exponential decay of the waves toward the edges of plate B produced by the damping can also be seen. The response in the shadow zone shown in Figure 3.6 is negative, opposite that shown in Figure 3.1. Again with light damping, plate B is excited both in and out of the shadow zone, as shown in Figure 3.8.

Figure 3.9 is a surface plot of the inverse Fourier transform of equation 2.19 using the same input values as for Figure 3.1 and 3.2. This is the pressure incident on plate B at a low resonance frequency and high damping (37.9 Hz and η=0.01). It can be seen in this figure that the pressure distribution on plate B closely follows the mode shape of plate A as shown in Figure 3.1. Figure 3.10 is a surface plot of the inverse Fourier transform of equation 2.25 using the same inputs. This is the Green's function of Plate B. Equation 2.23 is the response of plate B and is obtained by combining equations 2.19 and 2.25 and multiplying by 2. Similarly, if each point on Figures 3.9 and 3.10 are combined and multiplyed by 2, the result is Figure 3.2, the response of plate B. Figures 3.11 and 3.12 are the pressure and Green's functions respectively for 110.0 Hz and η=0.1. These two figures can be combined in the same manner to obtain the plate response shown on Figure 3.6. Comparing the pressures shown in Figures 3.9 and 3.11, the effects of the resonant response of the plate on the increase in the pressure incident on plate B in the shadow zone can be seen. At resonance, the incident pressure is higher. Figures 3.10 and 3.12 show the effects of free propagation in the infinite plate outside the shadow zone. At the higher frequency, the wavelength for free propagation is smaller.

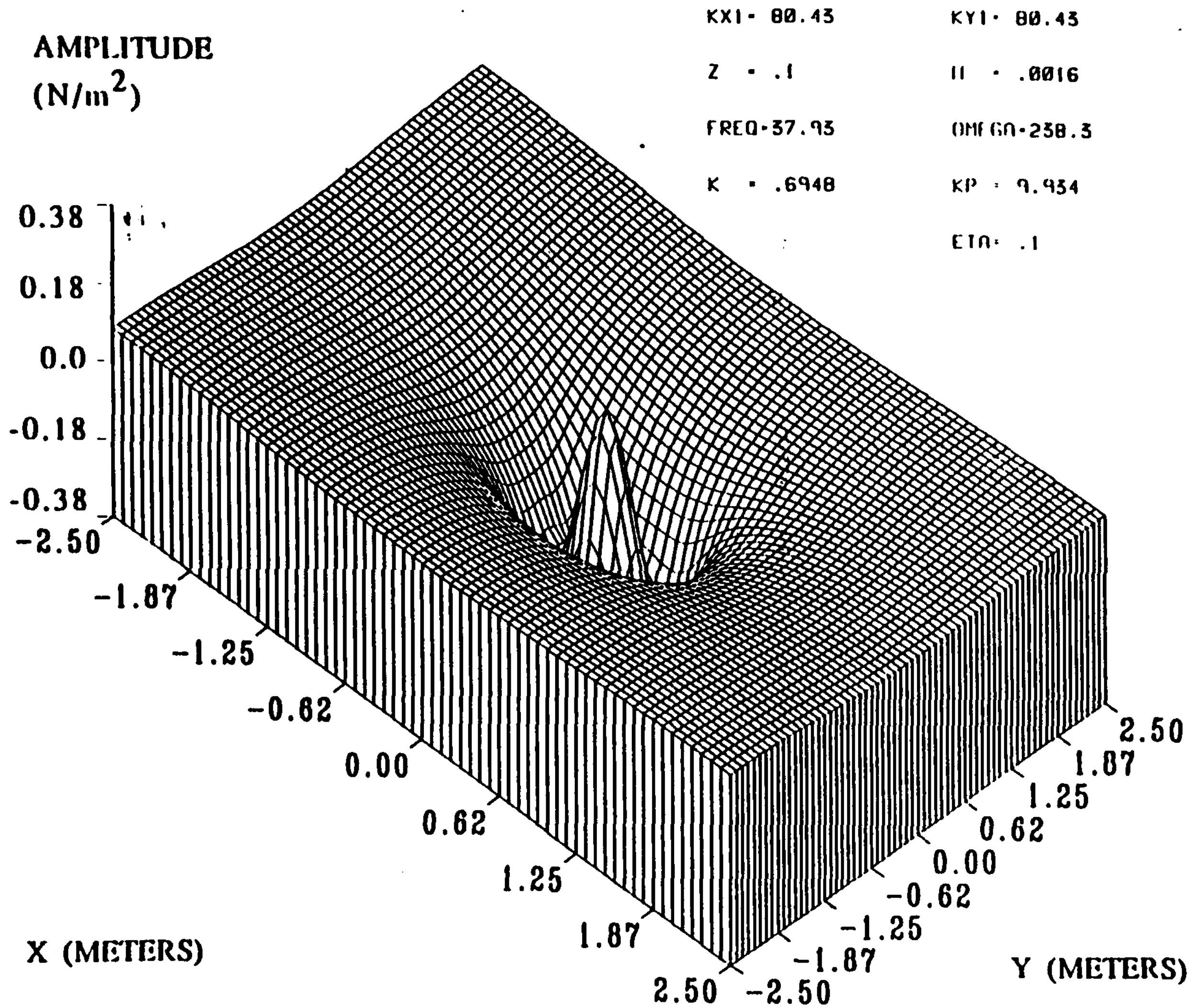

Figure 3.9 Pressure incident on plate B, low resonance, high damping (f=37.9Hz and $\eta = 0.1$)

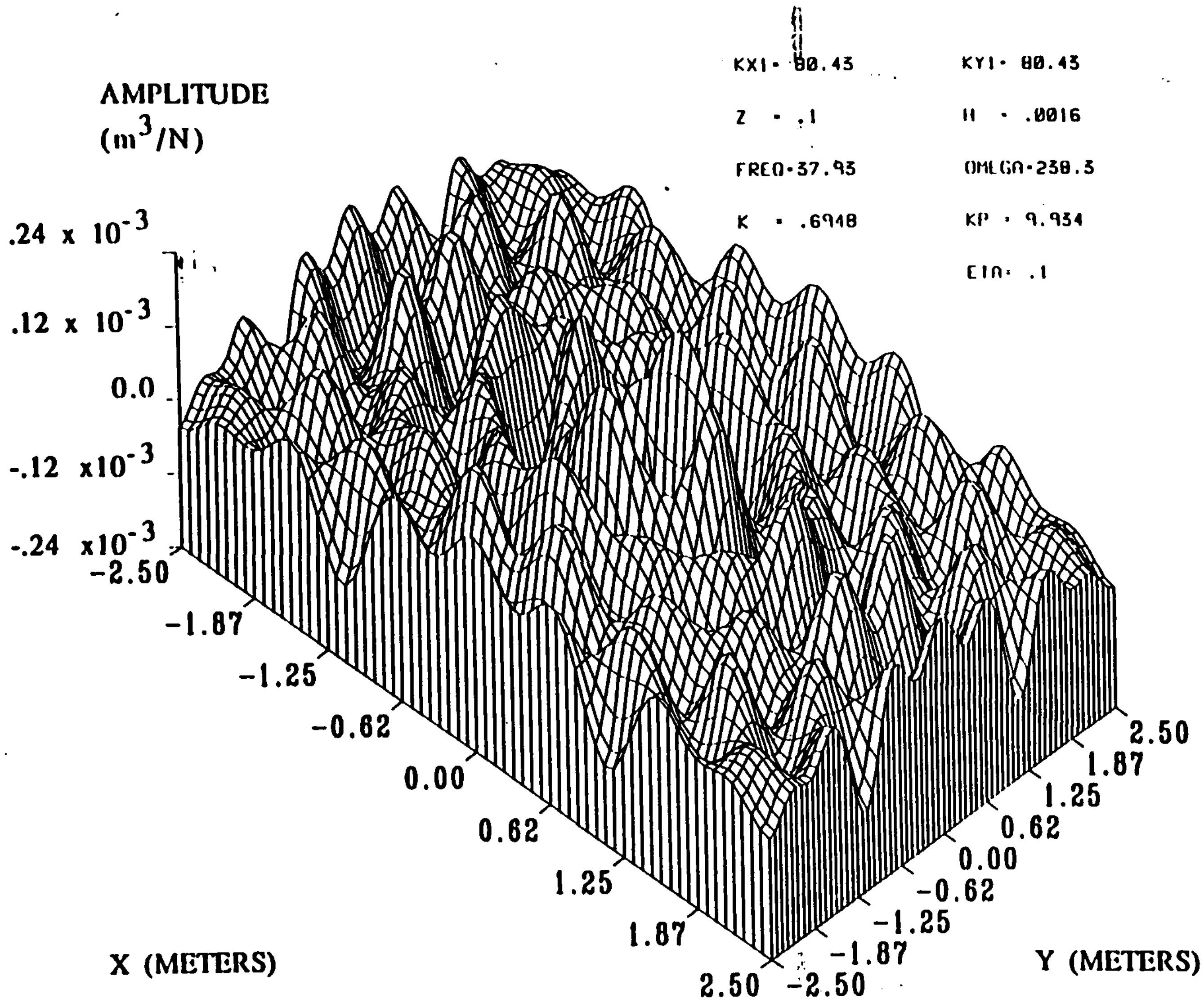

Figure 3.10 Green's function for plate B, low resonance, high damping (f=37.9Hz and $\eta = 0.1$)

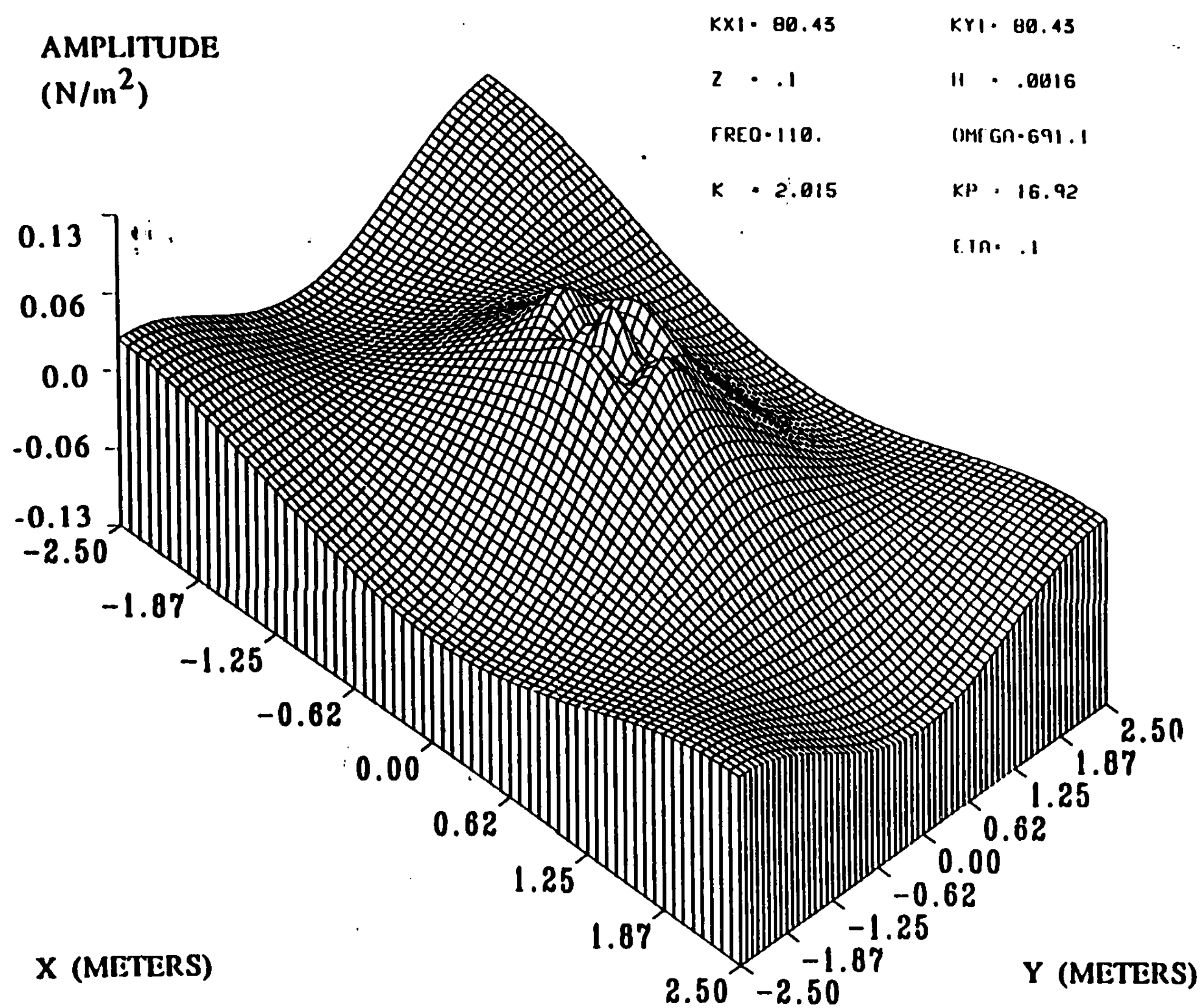

Figure 3.11 Pressure incident on plate B, low non-resonance, high damping (f=110.0Hz and $\eta = 0.1$)

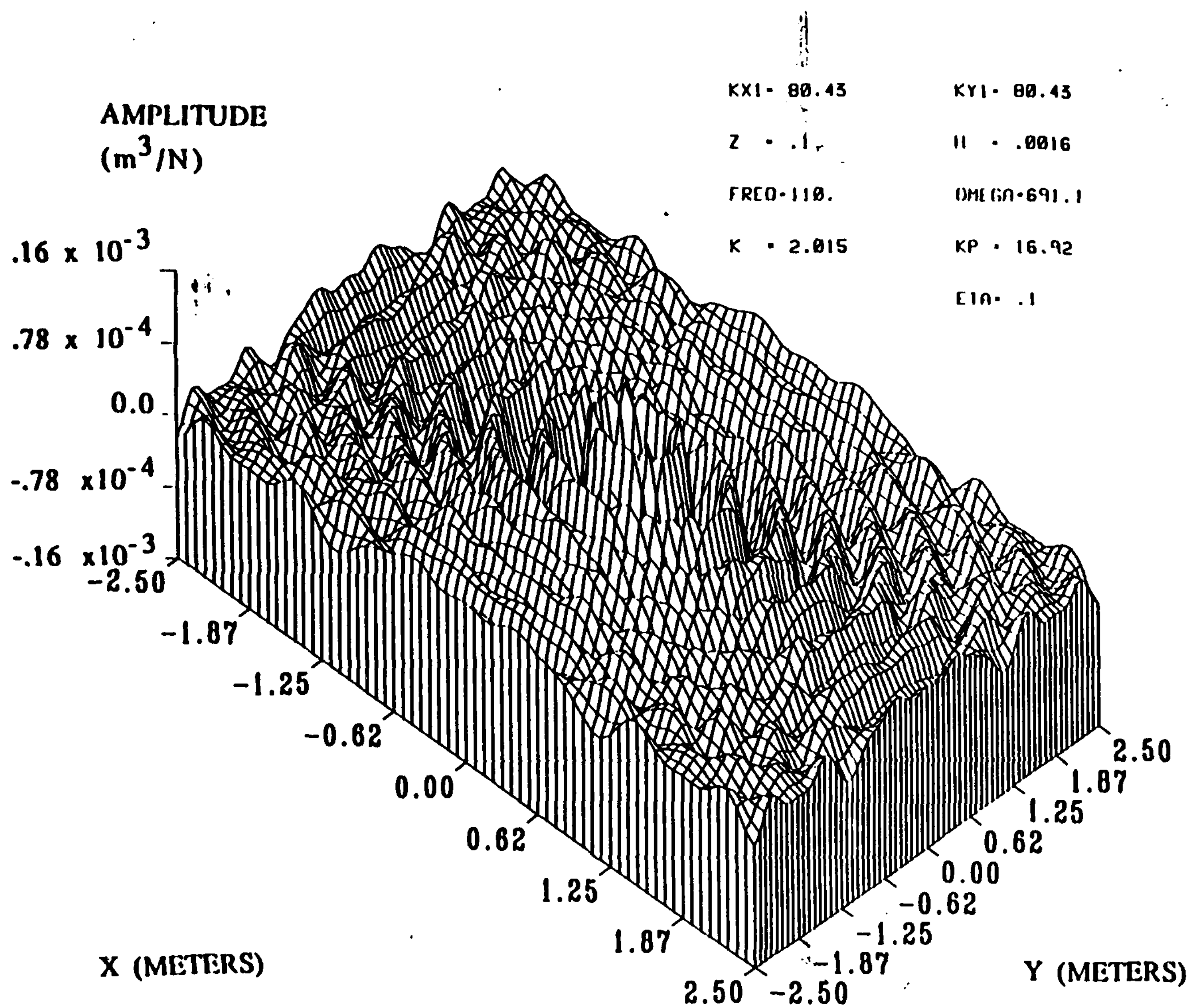

Figure 3.12 Green's function for plate B, low non-resonance, high damping (f=110.0Hz and η = 0.1)

3.2. Transmission Loss between plates A and B

Predictions of the motions of plate A and plate B obtained by using equations 2.9 and 2.28 in equation 2.40 to predict the transmission loss in decibels from 10 Hz to 5000 Hz are presented in Figures 3.13 through 3.18. Unless otherwise noted, the parameters listed in Table 3.1 are used, with the separation distance between the plates, the thickness of plate B, the size of plate A, and the damping varied. The values in Table 3.1 were chosen to simulate a typical sub-base and foundation for a medium sized piece of equipment. Also shown in Figures 3.13 through 3.18 is the theoretical transmission loss for simple resilient mounts with a resonance frequency of 5 Hz as computed in equation 2.1 and shown in Figure 2.2. The parameters given by Snowdon (2) for rubber mounts were used in equation 2.1.

In Figure 3.13 the effect on airborne transmission loss of changing the separation distance between the plates from 0.01 m to 0.5 meters is presented. Above 100 Hz all of the curves, except for the curve for 0.5 m, fall below the transmission loss curve for the mounts. This implies that the airborne path would degrade the performance of the isolation system even for the largest separation distances. Increasing the separation distance increases the airborne transmission losses at frequencies between 100 and 1K Hz. Below 100 Hz, the separation distance has little impact on the airborne transmission losses, other than at the frequencies where maxima and minima occur in the transmission loss. The minima in the transmission loss curve below 100 Hz may be due to a resonance associated with the effective masses of the plates and the effective stiffness of the air between the plates. For separation distances greater than 0.05 m, plate B moves out of the acoustic nearfield of plate A at frequencies above 1K Hz, so

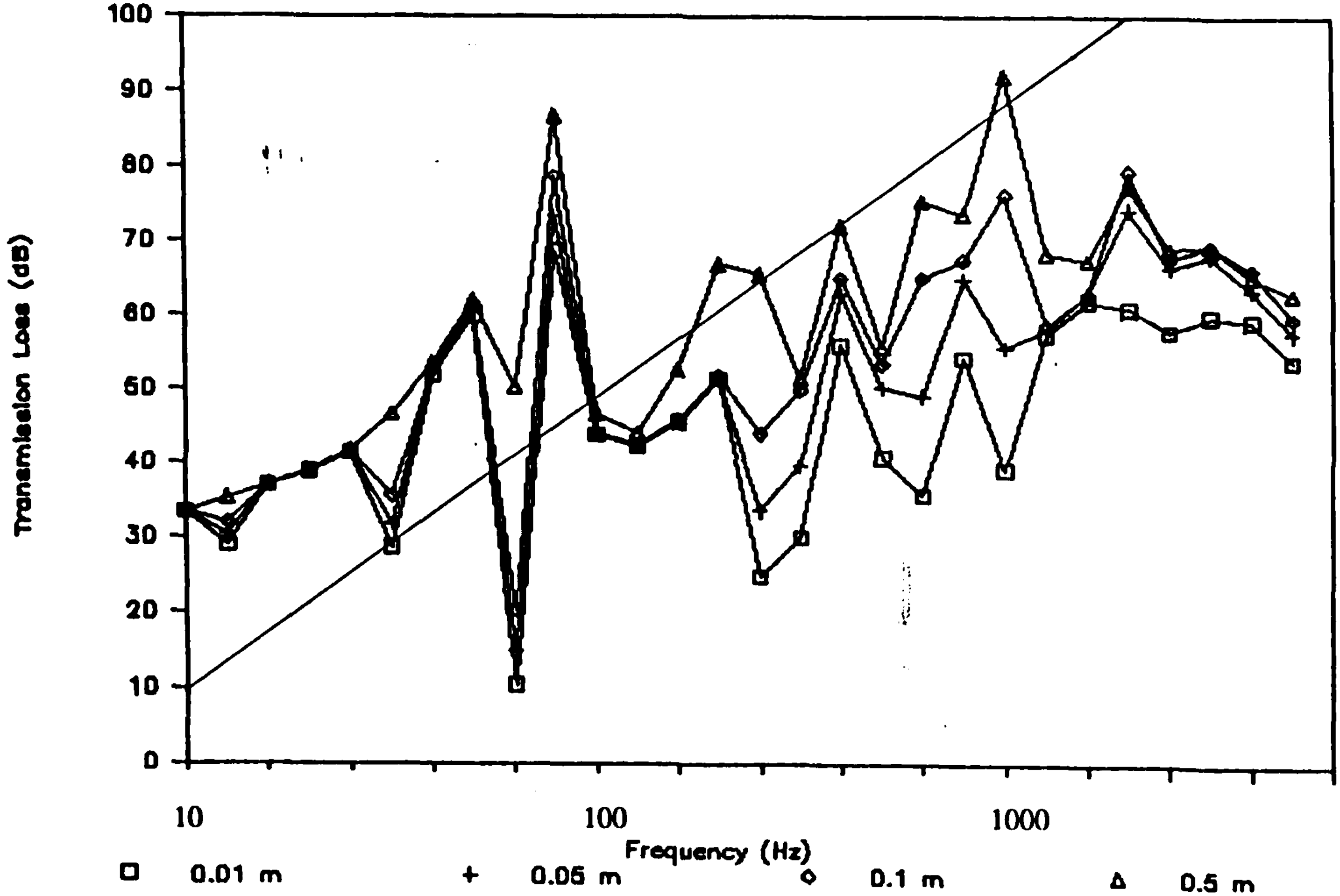

Figure 3.13 Transmission loss curves, changing separation distance between the plates

that the transmission losses above 1K Hz become less sensitive to separation distance than at frequencies below 1K Hz.

In Figure 3.14 the effect on airborne transmission of changing the thickness of plate B from 0.1587 cm (1/16 in.) to 1.27 cm (1/2 in.) is shown. Below 1K Hz, increasing the thickness of the receiver plate increases the airborne transmission losses at all frequencies. Above 100 Hz, the transmission losses for the 0.0157 cm (1/16 in.) thick plate are below the transmission losses for the mounts. Below 1K Hz, the transmission losses for the two thicker plates are above the losses for the mounts. Above 1K Hz, the transmission losses for all three plates are below the losses for the mounts. Increasing the stiffness of the receiving plate by increasing the thickness increases the airborne transmission loss so that the airborne path is not significant at lower frequencies (below 1K Hz); however, at higher frequencies (above 1K Hz) the airborne path remains a problem even when the stiffness of the structure is high.

In Figure 3.15 the size of plate A is changed from 1 m x 1 m to 0.1 m x 0.1 m. Above 100 Hz, the transmission losses for the two larger plates are below the transmission losses for the mounts and above 300 Hz, the airborne transmission losses for the smallest plate is also below the losses for the mounts. Decreasing the area of plate A without changing the separation distance increases the ratio of the area on the sides between the plates available for air to escape to the surface area of plate A and the surface of plate B in the shadow zone. Thus, with the smaller plate, a larger percentage of the energy in between the plates is permitted to escape which increases the airborne transmission losses between the plates at low frequencies. At the higher frequencies, air entrapment becomes less of a factor, decreasing the effect of the size of plate A on the airborne transmission loss. Also the dip and peak in the airborne transmission loss curve for the large (1 m x 1 m) plate are reduced when the size of plate A is decreased. This may be a damping effect produced by an increase in the percentage of the energy

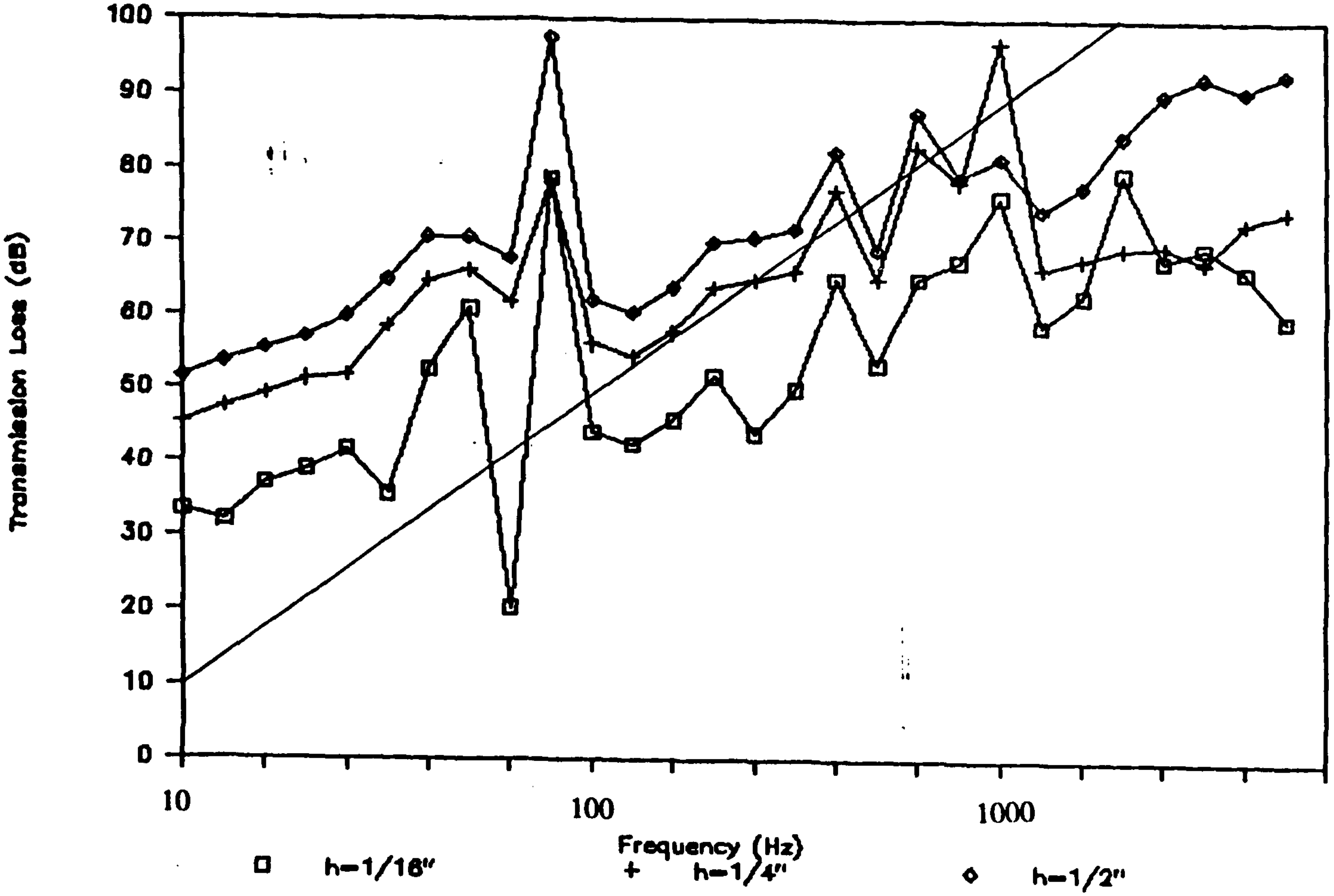

Figure 3.14 Transmission loss curves, changing the thickness of plate B

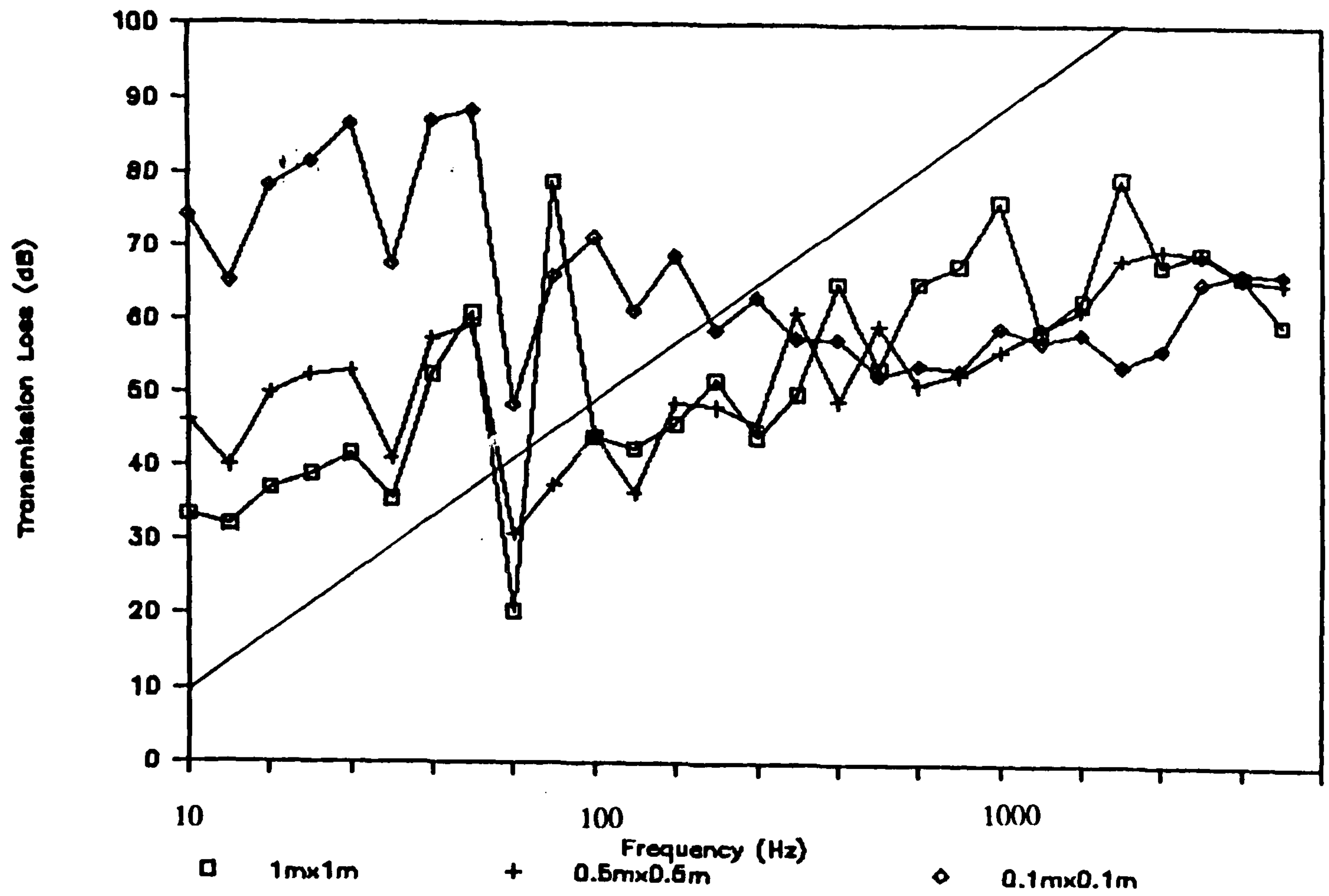

Figure 3.15 Transmission loss curves, changing the size of plate A

between the plates lost to propagation out from between the plates through the openings at the sides of the plates.

In Figures 3.16 and 3.17 the damping in plates A and B was changed for two different separation distances between the plates. In Figure 3.16 the distance between the plates is 0.1 m and the damping is increased from 0.001 (a typical number for an undamped steel plate) to 0.1 (a typical number for steel with damping applied). In Figure 3.17 the distance between the plates is 0.01 m and the damping is varied in the same manner. For both of these graphs the airborne transmission loss curve for the highly damped plate follows the same trend as for the lightly damped plate; however, with damping, the curves are much flatter. This shows the effect of damping which works primarily at resonances. For both separation distances, damping increases the airborne transmission losses at 63 Hz where the airborne transmission loss with light damping is below the transmission loss curve for the mounts. However, above 100 Hz, where the airborne transmission loss curve does not display the effects of resonances, damping has less effect on the airborne transmission losses.

Adding damping to only plate B produced the results shown in Figure 3.18 for a separation distance of 0.1 m. The damping in plate B has an effect only at the frequencies below 100 Hz where a peak and a dip in the airborne transmission loss curve occurs. Again the damping appears to damp the resonant effect that occurs between the plates and the air cushion between the plates.

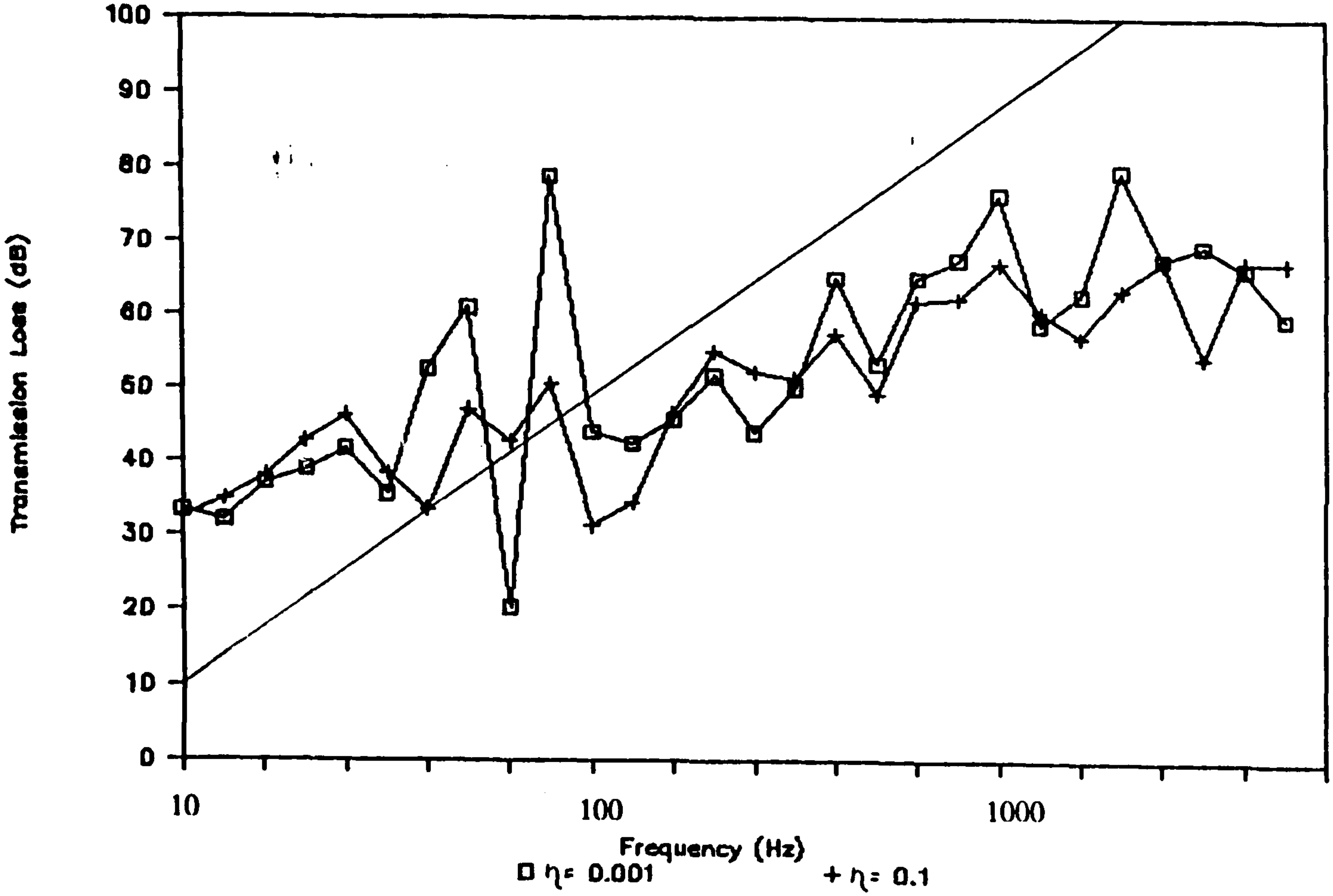

Figure 3.16 Transmission loss curves, changing the damping in plates A and B for a plate separation distance of 0.1 m

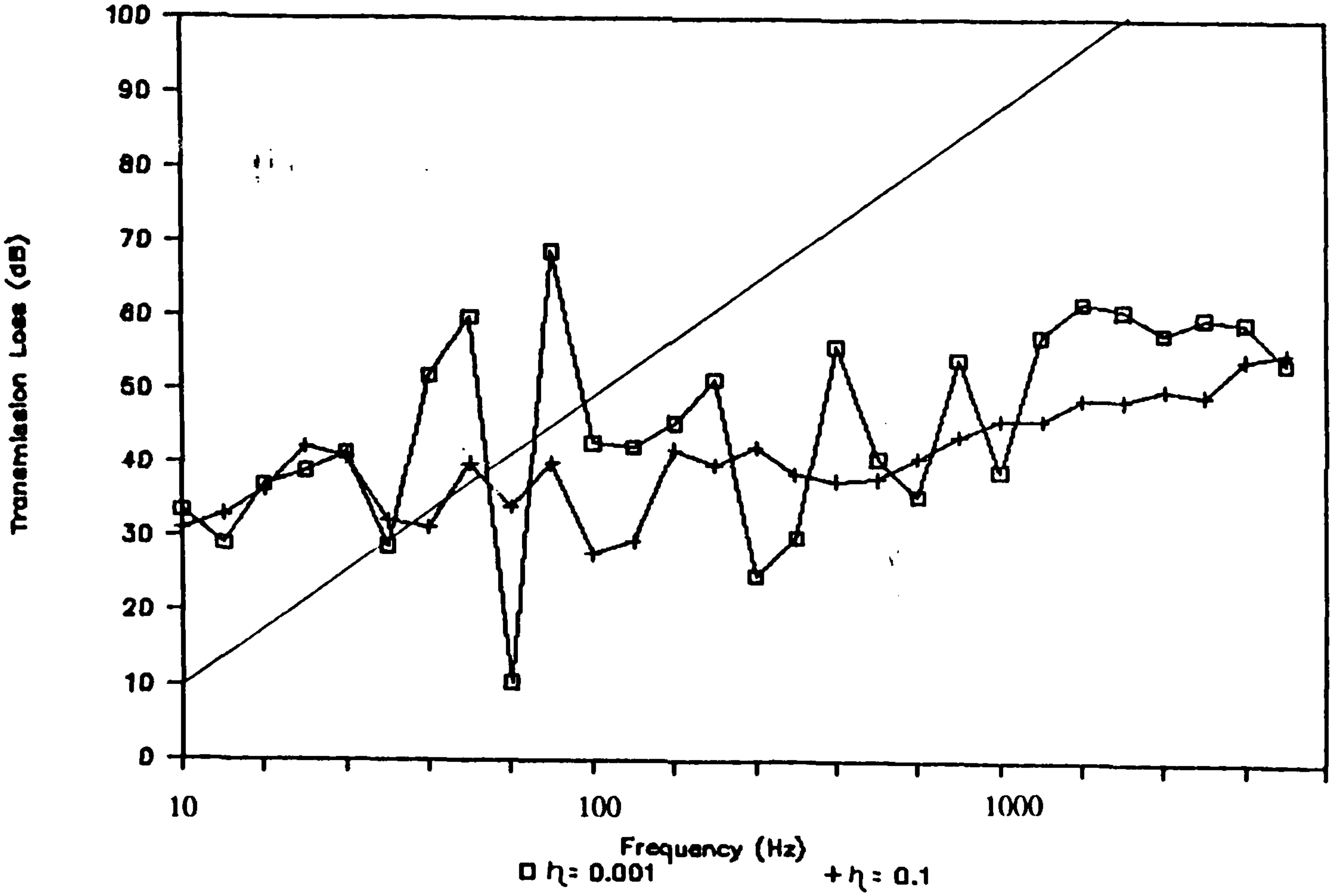

Figure 3.17 Transmission loss curves, changing the damping in plates A and B for a plate separation distance of 0.01 m

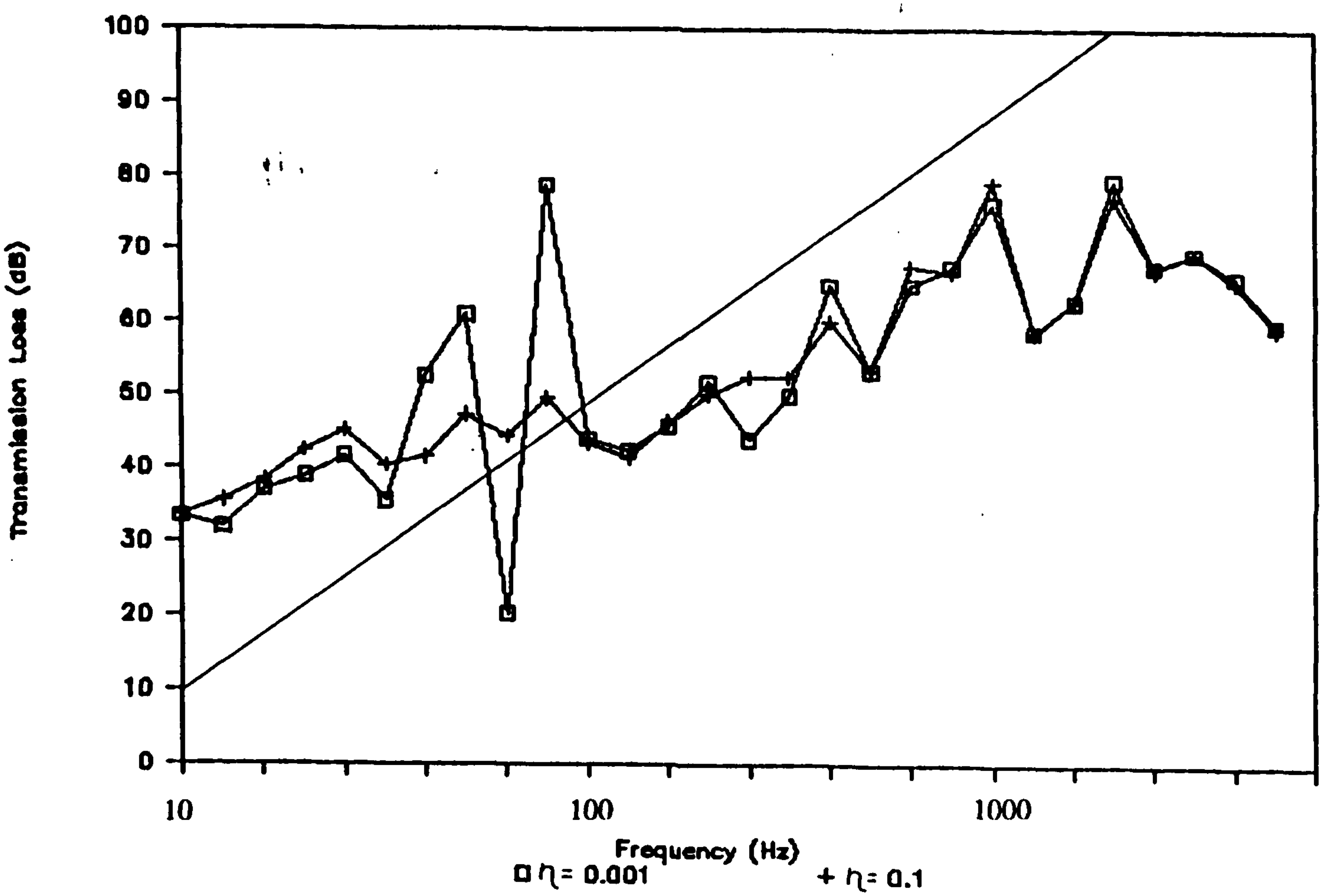

Figure 3.18 Transmission loss curves, changing the damping in plate B for a plate separation distance of 0.1 m

Chapter 4

CONCLUSIONS

An analytic model for the airborne transmission loss between a vibrating finite plate and a parallel infinite plate was developed. Comparison of the predicted airborne transmission losses with predicted transmission losses for resilient mounts revealed that airborne transmission losses were often less than the losses through the mounts at frequencies above 100 Hz, implying that the airborne transmission path may be significant in many well-designed single-stage resilient mounting systems. Below 100 Hz, the transmission losses through the mounts are below the airborne transmission losses, except at frequencies where resonances between the plates occur.

Increasing the separation distance between the plates, increasing the thickness of the receiver plate and decreasing the size of the source plate increased the airborne transmission losses between the plates. Increasing the damping in the plates was most effective in reducing peaks and dips in the airborne transmission loss curves below 100 Hz.

The basic assumptions used in developing the analytic model are that the impedances of the plates are much larger than the impedance of the air between the plates, there is no feedback from the receiver plate to the source plate and that the pressure generated in the plane, but outside the area, of the source plate has a negligible effect on the airborne transmission losses. In future research, these assumptions should be relaxed, particularly the feedback assumption where standing waves between the plates should decrease the airborne transmission losses. Also, relaxing the small air impedance assumption would increase the coupling between the air and the plates increasing the vibratory response of the plates and decreasing the airborne transmission losses. Experimental verification of the analytic model is needed to certify the conclusion

presented above. Also, experimental and theoretical analysis of treatments to increase the airborne noise transmission losses between structures, such as decoupling coating and damping, is needed.

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Appendix A

CALCULATION OF I_{mn}

In this appendix, the integral

$$I_{mn}(k_x,k_y) = \int_{-\infty}^{\infty}\int_{-\infty}^{\infty} \cos\left[\frac{(2m+1)\pi x}{L_x}\right]\cos\left[\frac{(2n+1)\pi y}{L_y}\right] e^{-ik_x x}\, e^{-ik_y y}\, dx\, dy \qquad \text{A.1}$$

as given in equation 2.11 is evaluated.

Since $\cos\left[\frac{(2m+1)\pi x}{L_x}\right]$ and $\cos\left[\frac{(2n+1)\pi y}{L_y}\right]$ are even and defined only for $-L_x/2 \le x \le L_x/2$ and $-L_y/2 \le y \le L_y/2$,

$$I_{mn}(k_x,k_y) = \int_{-L_x/2}^{L_x/2} \cos\left[\frac{(2m+1)\pi x}{L_x}\right] \cos(k_x x)dx \int_{-L_y/2}^{L_y/2} \cos\left[\frac{(2n+1)\pi y}{L_y}\right] \cos(k_y y)\, dy\,. \qquad \text{A.2}$$

When $\left[\frac{(2m+1)\pi x}{L_x}\right]^2 \neq k_x^2$ and $\left[\frac{(2n+1)\pi y}{L_y}\right]^2 \neq k_y^2$, equation A.2 becomes

$$I_{mn}(k_x,k_y) = \left\{ \frac{\sin\left[\frac{(2m+1)\pi x}{L_x} - k_x x\right]}{2\left[\frac{(2m+1)\pi}{L_x} - k_x\right]} + \frac{\sin\left[\frac{(2m+1)\pi x}{L_x} + k_x x\right]}{2\left[\frac{(2m+1)\pi}{L_x} + k_x\right]} \right\}_{-L_x/2}^{L_x/2}$$

$$* \left\{ \frac{\sin\left[\frac{(2n+1)\pi y}{L_y} - k_y y\right]}{2\left[\frac{(2n+1)\pi}{L_y} - k_y\right]} + \frac{\sin\left[\frac{(2n+1)\pi y}{L_y} + k_y y\right]}{2\left[\frac{(2n+1)\pi}{L_y} + k_y\right]} \right\}_{-L_y/2}^{L_y/2} . \qquad \text{A.3}$$

Using $\frac{\sin(\alpha)}{\alpha} = \frac{-\sin(-\alpha)}{\alpha}$,

$$\sin\left[\frac{(2m+1)\pi}{2} - \frac{k_x L_x}{2}\right] = (-1)^m \cos(\frac{k_x L_x}{2})$$

and

$$\sin\left[\frac{(2m+1)\pi}{2} + \frac{k_x L_x}{2}\right] = (-1)^m \cos(\frac{k_x L_x}{2})$$

in equation A.3,

$$I_{mn}(k_x,k_y) = \frac{4(-1)^{m+n} \frac{(2m+1)\pi}{L_x} \frac{(2n+1)\pi}{L_x} \cos\frac{k_x L_x}{2} \cos\frac{k_y L_y}{2}}{\left\{ \left[\frac{(2m+1)\pi}{L_x}\right]^2 - k_x^2 \right\} \left\{ \left[\frac{(2n+1)\pi}{L_y}\right]^2 - k_y^2 \right\}} \quad . \qquad \text{A.4}$$

When $\left[\frac{(2m+1)\pi x}{L_x}\right]^2 = k_x^2$ and $\left[\frac{(2n+1)\pi y}{L_y}\right]^2 \neq k_y^2$, equation A.2 becomes

$$I_{mn}(k_x,k_y) = \int_{-L_x/2}^{L_x/2} \cos^2\left[\frac{(2m+1)\pi x}{L_x}\right] dx \int_{-L_y/2}^{L_y/2} \cos\left[\frac{(2n+1)\pi y}{L_y}\right] \cos(k_y y)\, dy \, . \qquad \text{A.5}$$

Integration of equation A.5 yields

$$I_{mn}(k_x,k_y) = \frac{L_x (-1)^n \frac{(2n+1)\pi}{L_y} \cos\frac{k_y L_y}{2}}{\left[\frac{(2n+1)\pi}{L_y}\right]^2 - k_y^2} \quad . \qquad \text{A.6}$$

Similarly when $\left[\frac{(2m+1)\pi x}{L_x}\right]^2 \neq k_x^2$ and $\left[\frac{(2n+1)\pi y}{L_y}\right]^2 = k_y^2$, equation A.2 becomes

$$I_{mn}(k_x,k_y) = \frac{L_y\,(-1)^m \frac{(2m+1)\pi}{L_x} \cos\frac{k_x L_x}{2}}{\left[\frac{(2m+1)\pi}{L_x}\right]^2 - k_x^2} \quad . \tag{A.7}$$

For $\left[\frac{(2m+1)\pi x}{L_x}\right]^2 = k_x^2$ and $\left[\frac{(2n+1)\pi y}{L_y}\right]^2 = k_y^2$, equation A.2 becomes

$$I_{mn}(k_x,k_y) = \int_{-L_x/2}^{L_x/2} \cos^2\left[\frac{(2m+1)\pi x}{L_x}\right] dx \int_{-L_y/2}^{L_y/2} \cos^2\left[\frac{(2n+1)\pi y}{L_y}\right] dy \,, \tag{A.8}$$

which, when integrated, yields

$$I_{mn}(k_x,k_y) = \frac{L_x\,L_y}{4} \quad . \tag{A.9}$$

Appendix B

COMPUTER MODEL

In order to solve the equation of motion of both plate A and plate B and to compute the transmission loss between the two plates, three separate computer programs were developed.

Program PRA (Plate Response of plate A) computes the mode shape of plate A. Program PR (Plate Response) computes the response of plate B. Program PRTL (Plate Response - Transmission Loss) computes the transmission loss between plate A and plate B. PRTL is broken down into three parts for three different frequency ranges: PRTL100 for the frequency range from 10 to 100 Hz, PRTL1000 for the frequency range from 100 to 1000 Hz and PRTL5K for the frequency range from 1000 to 5000 Hz. The source code for each of these programs is in Appendix C.

The inputs to all of these programs are taken from the same input file format. A sample input and its corresponding input variables are included in Appendix C. Not all of these inputs are used in each of the programs. See the corresponding write up below for which variables are needed for each program.

B.1. Program PRA

This program solves equation 2.9 to obtain the mode shape of plate A. The inputs to this program are:

Lx and Ly	Dimensions of plate A
Xo and Yo	The point where the input force is applied to plate A
ROA	Density of plate A
ACHA	Thickness of plate A
ETAA	Damping coefficient for plate A
EA	Modulus of Elasticity of plate A

The output from this program is a surface plot of the mode shape of plate A. The plotting program used to obtain these plots is a subroutine called ARL_HIDE and is available on the ARL Penn State VAX computer.

B.2. Program PR

This program uses equation 2.28 to obtain the vibration response of plate B to excitation by airborne transmission from plate A. The integral portion of equation 2.28 is solved using a 2-dimensional discrete inverse fast Fourier transform program called F2T2B in the IMSL subroutine library, also available on the ARL Penn State VAX computer. In equation 2.28 this is an infinite 2-dimensional integral; however, in order to solve this integral using the computer the finite integral is approximated by a finite series using equation 2.38.

The choice of wavenumber range (KX1 and KY1) and the sample rate determines the size of plate B by the following equations,

$$x = \frac{2N}{KX1} \qquad y = \frac{2N}{KY1} \qquad \text{B.1}$$

where x and y are the dimensions of plate B.

The wavenumber range must be chosen judiciously with respect to the frequency of interest because of the relationship of the wavenumber and frequency in equation 2.28. At lower frequencies a lower wavenumber range can be used, therefore a larger area of plate B can be observed using equation B.1. At higher frequencies a higher wavenumber range must be used and a smaller area of plate B can be observed using the

same sampling rate (N). A higher sampling rate will solve this problem however, the computer time also increases.

The inputs to this program are:

Lx and Ly	Dimensions of plate A
Xo and Yo	The point where the input force is applied to plate A
ROA	Density of plate A
ROB	Density of plate B
ACHA	Thickness of plate A
ACHB	Thickness of plate B
ETAA	Damping coefficient for plate A
ETAB	Damping coefficient for plate B
EA	Modulus of Elasticity of plate A
EB	Modulus of Elasticity of plate B
Z	Distance between the plates
PRA	Prandel number for the fluid
GAMMA	Specific heat ratio for the fluid
MU	Viscosity of the fluid
NU	Poison's ratio
C	Speed of sound in the fluid
RO	Density of the fluid
KX1	Upper wavenumber limit in the x direction
KY1	Upper wavenumber limit in the y direction

The output of this program is a surface plot of the vibration response of plate B. ARL_HIDE is also used to obtain these plots.

B.3. Program PRTL

Program PRTL uses the same algorithms used in programs PRA and PR to obtain the motions of plates A and B. Equation 2.40 is then used to obtain the transmission loss between the two plates in the shadow zone of plate A.

This program has been broken down into three parts; PRTL100, PRTL1000 and PRTL5K in order to change the sampling rate and wavenumber range for the frequency range covered by each PRTL program.

The input to these programs are the same as for program PR.

The output from these programs is a table of transmission losses as a function of frequency.

Appendix C

FORTRAN COMPUTER PROGRAMS

```
      PROGRAM PRA
C  PLATE RESPONSE OF PLATE-A, FINITE PLATE
C  WRITTEN BY MICHAEL F SHAW
C
      INTEGER OMEGAI, OMEGAB, OMEGAT, INC, EM, EN, N, I, J,
     &N1,N2,N11
C  NSHOULD BE A PRODUCT OF SMALL PRIMES
      PARAMETER (N=128)
      COMPLEX  DSTAR,OMEGAMNSQ,B(N,N),BT(N,N),WA(N,N),WA2(N,N),
     &BT2(N,N)
      REAL VMAX,VMIN,VINC,WAREAL(N+1,N+1),WORK1(4000),WORK2(100)
      REAL DA,ETAA,ETAB,LX,LY,ROA,ROB,ACHA,ACHB,WA2REAL(N+1,N+1),
     &KX1,KY1,F,MP,XO,YO,NU,PI,OMEGA,GAMMA,RO,MU,C,EA,EB,Z,
     &OMEGA1,OMEGA2,OMEGAMN0,OMEGAMN1,OMEGAMN2
      INTRINSIC CMPLX
C  ****************************************************************
C                    VARIABLE LIST
C
C   DA - BENDING RIGIDITY OF THE FINITE PLATE
C   ETAA - DAMPING COEFICIENT FOR THE FINITE PLATE
C   LX & LY - DIMENSIONS OF THE FINITE PLATE
C   ROA - DENSITY OF THE FINITE PLATE
C   ACHA - THICKNESS OF THE FINITE PLATE
C   RO - DENSITY OF FLUID
C   XO & YO - THE POINT WHERE THE INPUT FORCE F IS APPLIED
C   NU - POISON'S RATIO
C   EA - SHEER MODULUS OF THE FINITE PLATE
C  ****************************************************************
      PARAMETER (F=1.0)
C   INPUT FORCE TO THE FINITE PLATE IS ASSUMED TO BE A UNIT FORCE (1)
      OPEN (UNIT=25, FILE='PR.DAT', STATUS='OLD')
      OPEN (UNIT=26, FILE='PRA.OUT', STATUS='NEW')
C  READ INPUT DATA FROM PR.DAT
      READ (25,15) LX,LY,XO,YO,ACHA,ACHB
 15   FORMAT(4F8.3,2F8.6)
      READ (25,25) ROA,ROB,RO,ETAA,ETAB,GAMMA
 25   FORMAT(2F8.2,F8.3,2F8.7,F8.6)
      READ (25,35) MU,C,EA,EB,KX1,KY1
 35   FORMAT(F8.7,F8.3,2E8.3,2F8.2)
      READ (25,45) NU,OMEGA1,OMEGA2,INC,Z,OMEGA
 45   FORMAT(F8.4,F8.1,F16.1,I8,F8.3,F12.3)
C  COMPUTE THE VALUE OF PI
      PI=2.0*ASIN(1.0)
C  COMPUT THE BENDING RIGIDITY
      DA=(EA*ACHA**3.0)/(12.0*(1.0-NU**2.0))
      DSTAR=CMPLX(DA,DA*ETAA)
      MP=ROA*ACHA*LX*LY
      WRITE(26,11) LX,LY
 11   FORMAT(' PARAMETERS OF FINITE PLATE',/,' DIMENSIONS OF PLATE'
     &,' (LX)   ',F8.3,' (LY) ',F8.3,'  METERS')
      WRITE(26,21) XO,YO
 21   FORMAT(' DRIVING POINT   XO ',F8.3,' YO ',F8.3,'
     & METERS')
      WRITE(26,31) F
 31   FORMAT(' DRIVING FORCE   ',F8.3,'  N')
      WRITE(26,41)ACHA
 41   FORMAT(' PLATE THICKNESS   ',F8.6,'  M')
      WRITE(26,51) ROA
 51   FORMAT(' PLATE DENSITY   ',F8.3,'  KG/CU-METERS')
      WRITE(26,61) EA
 61   FORMAT(' MODULUS  OF ELASTICITY ',E8.3,' N/SQ-METERS')
      WRITE(26,71) DA
 71   FORMAT(' BENDING RIGIDITY  ',E12.4)
      WRITE(26,81) ETAA
 81   FORMAT(' DAMPING COEFFICIENT',F8.4)
C    THE FREQUENCY RANGE IS TAKEN FROM OMEGA1 TO OMEGA2 IN INCREMENTS
```

```
C     CORRESPONDING TO INC
C
C        OMEGAB=10*IFIX(ALOG10(OMEGA1))
C        OMEGAT=INC*10*IFIX(ALOG10(OMEGA2))
C        DO 1000 OMEGAI=OMEGAB,OMEGAT
C        OMEGA=2.0*PI*(10.0**(OMEGAI/(INC*10.0)))
          DO 100 I=1,N
          DO 100 J=1,N
          BT(I,J)=0.0
          WAREAL(I,J)=0.0
100       CONTINUE
        WRITE(26,12) OMEGA/(2.0*PI)
12      FORMAT(/,' FREQUENCY  ',F8.3,'   HZ')
C  THE NEXT SECTION OF THE PROGRAM UP TO LINE 58 COMPUTES THE UPPER
C  AND LOWER MODES WHICH THE PLATE RESPONSE IS SUMMED OVER. N1 IS
C  THE LOWER LIMIT AND N2 IS THE UPPER LIMIT
C  INITIALIZE AND INCREMENT N1
        N1=-1
18      N1=N1+1
C  CALCULATE RESONANT OMEGA FOR EN=N1, EM=N1 MODE
        OMEGAMN0=(DA/(ROA*ACHA))**.5*(((2.0*N1+1.0)*PI/LX)**2.0
       &+((2.0*N1+1.0)*PI/LY)**2.0)
C  IF RESONANT OMEGA IS GREATER THAN OMEGA GO ON TO 28 IF NOT
C  INCREMENT N1 AND REPEAT
        IF (OMEGAMN0.GT.OMEGA) GOTO 28
        GOTO 18
C  INITIALIZE AND INCREMENT N2
28      N2=-1
38      N2=N2+1
C  CALCULATE THE RESONANT OMEGAS FOR EN=N2, EM=0 AND EN=0, EM=N2
C  MODES AND TEST IF LARGER THAN OMEGA
        OMEGAMN1=(DA/(ROA*ACHA))**.5*(((2.0*N2+1.0)*PI/LX)**2.0
       &+(PI/LY)**2.0)
        OMEGAMN2=(DA/(ROA*ACHA))**.5*((PI/LX)**2.0
       &+((2.0*N2+1.0)*PI/LY)**2.0)
C  IF LARGER THAN OMEGA GO ON TO 48 IF NOT INCREMENT N2 AND REPEAT
        IF((OMEGAMN1.GT.OMEGA).OR.(OMEGAMN2.GT.OMEGA)) GOTO 48
        GOTO 38
48      CONTINUE
C  ADD THREE MODES TO UPPER LIMIT
        N2=N2+3
        N11=N1
C  SUBTRACT THREE MODES FROM LOWER LIMIT, ZERO IS LOWEST N1
        N1=N1-3
        IF(N11.EQ.0) N1=0
        IF(N11.EQ.1) N1=0
        IF(N11.EQ.2) N1=0
        WRITE (6,58) N1,N2
58      FORMAT (2I5)
C  THE PLATE MODES ARE TAKEN FROM EM, EN = N1 TO N2.
        DO 200 EM=N1,N2
        DO 200 EN=N1,N2
          DO 300 I=1,N
          DO 300 J=1,N
          BT2(I,J)=BT(I,J)
300       CONTINUE
C  OMEGAMNSQ IS THE RESONANT OMEGA**2.0
        OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
       &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
          DO 400 I=1,N
          DO 400 J=1,N
          X=(-LX/2.0)+(LX*(I-1))/N
          Y=(-LY/2.0)+(LY*(J-1))/N
          B(I,J)=(COS((2.0*EM+1.0)*PI*XO/LX))*(COS((2.0*EN+1.0)
       &*PI*YO/LY))*(COS((2.0*EM+1.0)*PI*X/LX))*(COS((2.0*EN+1.0)
       &*PI*Y/LY))/(OMEGAMNSQ-OMEGA**2.0)
```

```
          BT(I,J)=BT(I,J)+B(I,J)
 400      CONTINUE
 200    CONTINUE
        WRITE (26,301)
 301    FORMAT (' PLATE RESPONSE')
          DO 500 I=1,N
          DO 500 J=1,N
          WA(I,J)=(4.0*F/MP)*BT(I,J)
          WAREAL(I,J)=WA(I,J)
          WA2(I,J)=(4.0*F/MP)*BT2(I,J)
          WA2REAL(I,J)=WA2(I,J)
C  WA IS THE COMPLEX PLATE RESPONSE
C  WAREAL IS THE REAL PART OF THE PLATE RESPONSE
C  WA2 IS THE COMPLEX PLATE RESPONSE OF THE PREVIOUS MODE
C  WA2REAL IS THE REAL PART OF THE PREVIOUS MODE
 500      CONTINUE
C  THE FOLLOWING DO LOOP ADDS ON THE FINAL POINT TO MAKE
C  THE MATRIX SYMETRICAL
          DO 550 I=1,N
          WAREAL(N+1,I)=0.0
          WAREAL(I,N+1)=0.0
          WA2REAL(N+1,I)=0.0
          WA2REAL(I,N+1)=0.0
 550      CONTINUE
C  OUTPUT PLATE RESPONSE
          DO 800 I=1,N+1
          DO 800 J=1,N+1
          WRITE (26,401) I-1,J-1,WA2REAL(I,J),I-1,J-1,WAREAL(I,J)
 401      FORMAT (' WA2REAL(',I4,',',I4,') = ',E12.4,'   WAREAL(',
     &I4,',',I4,') = ',E12.4)
C  CALCULATE THE MAXIMUM, MINIMUM, AND INCREMENT FOR THE PLOT
          VMAX=AMAX1(ABS(WAREAL(I,J)),ABS(VMAX))
          VMIN=-VMAX
          VINC=VMAX/2.0
 800      CONTINUE
C  PLOTTER CREATES A 3-D PLOT OF THE PLATE RESPONSE
        CALL PLOTTER (VMAX,VMIN,VINC,WAREAL,WORK1,WORK2,N,
     &OMEGA,ACHA,ETAA,LX,LY)
C1000   CONTINUE
        END
C
        SUBROUTINE PLOTTER (VMAX,VMIN,VINC,WAREAL,WORK1,WORK2,N,
     &OMEGA,ACHA,ETAA,LX,LY)
C PLOTTER USES THE ARL_HIDE/TEMPLATE PLOTTING PACKAGE TO CREATE
C  3-D PLOTS OF THE PLATE RESPONSE
        REAL TXMIN,TYMIN,TXMAX,TYMAX,TXINC,TYINC
     &OMEGA,ACHA,ETAA,PI,LX,LY
        PI=2.0*ASIN(1.0)
        TXMIN=-LX/2.0
        TYMIN=-LY/2.0
        TXMAX=LX/2.0
        TYMAX=LY/2.0
        TXINC=LX/8.0
        TYINC=LY/8.0
        WRITE (6,501) TXMIN,TXMAX,TXINC,TYMIN,TYMAX,
     &TYINC,N
501     FORMAT (3F12.4,/,3F12.4,I4)
        CALL USLPDF
        CALL UPSET( 'OUTPUTFILE',7.0)
        CALL UASSGN(7.0, 'PRA.PDF\')
        CALL USTART
        CALL UDIMEN( 9.0,6.5)
        CALL USET('PERCENT')
        CALL UFONT('CROM')
        CALL USET('LARGE')
        CALL UPRINT(61.,83.,'FREQ=\')
```

```
      CALL UPRNT1(OMEGA/(2.0*PI),'REAL')
      CALL UPRINT(77.,83.,'OMEGA=\')
      CALL UPRNT1(OMEGA,'REAL')
      CALL UPRINT(61.,73.,'H  = \')
      CALL UPRNT1(ACHA,'REAL')
      CALL UPRINT(77.,73.,'ETA= \')
      CALL UPRNT1(ETAA,'REAL')
      CALL ARL_HIDE_FONT
      CALL UFONT('CROM')
      CALL USET('LARGE')
      CALL ARL_HIDE_SCALE(N+1,N+1,N+1,4000,N+1,VMIN,VMAX,
     &'CUTOFF',CUTOFF,0.5,0.6,1.25,0.5,0.5,0.0,45.0,45.0,
     &'FDRAW','SDRAW')
      CALL ARL_HIDE(WAREAL,WORK1,WORK2)
      CALL ARL_HIDE_FAXIS('VIEW','REAL',TXMIN,TXMAX,TXINC,
     &'(F8.2)','X  METERS',1,50.0,50.0)
      CALL ARL_HIDE_SAXIS('VIEW','REAL',TYMIN,TYMAX,TYINC,
     &'(F8.2)','Y  METERS',1,50.0,50.0)
      CALL ARL_HIDE_VAXIS('VIEW','REAL',VMIN,VMAX,VINC,
     &'(E12.4)','AMPLITUDE',1,50.0,50.0)
      CALL UEND
      RETURN
      END
```

```
      PROGRAM PR
C DETERMINES THE PLATE RESPONSE OF PLATE-B USING IMSL
C FFT SUBROUTINE
C
C WRITTEN BY MICHAEL F. SHAW
C
      INTEGER OMEGAI, OMEGAB, OMEGAT, INC, EM, EN, N, I, J,
     &NRCOEF,NCCOEF,LDCOEF,LDA,NLO,NUP,N11
      PARAMETER (N=128)
      COMPLEX DSTAR,OMEGAMNSQ,COEF(N,N),A(N,N),A1(N,N),
     &AT(N,N),KPF,KC,W(N,N),W1(N,N),AT1(N,N),WC(N,N),
     &TEMP
      REAL WFF1(4*N+15),WFF2(4*N+15),CPY(N,N)
      REAL VMAX,VMIN,VINC,WREAL(N+1,N+1),WORK1(4000),WORK2(100)
      REAL DA,DB,ETAA,ETAB,LX,LY,ROA,ROB,ACHA,ACHB,KX(N),KY(N),
     &KX1,KY1,KXM,KYN,F,RO,MP,XO,YO,LMN,Z,PRA,GAMMA,MU,NU,C,E,
     &HKX,HKY,PI,K,KP,OMEGA,OMEGA1,OMEGA2,OMEGAMNO,
     &OMEGAMN1,OMEGAMN2,W1REAL(N+1,N+1),TEMP2,TEMP1
      INTRINSIC CMPLX
      EXTERNAL F2T2B,FFTCI
C THE NEXT TWO LINES ALLOW N TO BE LARGER THAN 64 POINTS
      COMMON /WORKSP/ RWKSP
      REAL RWKSP(8884)
C ****************************************************************
C                      VARIABLE LIST
C
C  DA - BENDING RIGIDITY OF THE FINITE PLATE
C  DB - BENDING RIGIDITY OF THE INFINITE PLATE
C  ETAA - DAMPING COEFICIENT FOR THE FINITE PLATE
C  ETAB - DAMPING COEFICIENT FOR THE INFINITE PLATE
C  LX & LY - DIMENSIONS OF THE FINITE PLATE
C  ROA - DENSITY OF THE FINITE PLATE
C  ROB - DENSITY OF THE INFINITE PLATE
C  ACHA - THICKNESS OF THE FINITE PLATE
C  ACHB - THICKNESS OF THE INFINITE PLATE
C  RO - DENSITY OF FLUID
C  XO & YO - THE POINT WHERE THE INPUT FORCE F IS APPLIED
C  Z - DISTANCE BETWEEN THE PLATES
C  PRA - PRANDEL NUMBER FOR FLUID
C  GAMMA - SPECIFIC HEAT RATIO FOR FLUID
C  MU - VISCOSITY OF FLUID
C  NU - POISON'S RATIO
C  C - SPEED OF SOUND IN FLUID
C  EA - SHEER MODULUS OF THE FINITE PLATE
C  EB - SHEER MODULUS OF THE INFINITE PLATE
C  KX1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN X DIR
C  KY1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN Y DIR
C ****************************************************************
      PARAMETER (F=1.0)
C  INPUT FORCE TO THE FINITE PLATE IS ASSUMED TO BE A UNIT FORCE (1)
      OPEN (UNIT=25, FILE='PR.DAT', STATUS='OLD')
      OPEN (UNIT=26, FILE='PR.OUT', STATUS='NEW')
C ANOTHER LINE SO N CAN BE LARGER THAN 64
      CALL IWKIN(8884)
C READ INPUT DATA FROM PR.DAT
      READ (25,15) LX,LY,XO,YO,ACHA,ACHB
 15   FORMAT(4F8.3,2F8.6)
      READ (25,25) ROA,ROB,RO,ETAA,ETAB,GAMMA,PRA
 25   FORMAT(2F8.2,F8.3,2F8.7,2F8.6)
      READ (25,35) MU,C,EA,EB,KX1,KY1
 35   FORMAT(F8.7,F8.3,2E8.3,2F8.2)
      READ (25,45) NU,OMEGA1,OMEGA2,INC,Z,OMEGA
 45   FORMAT(F8.4,F8.1,F16.1,I8,F8.3,F12.3)
C COMPUTE THE VALUE OF PI
      PI=2.0*ASIN(1.0)
C COMPUTE THE BENDING RIGIDITIES
```

```
      DA=(EA*ACHA**3.0)/(12.0*(1.0-NU**2.0))
      DB=(EB*ACHB**3.0)/(12.0*(1.0-NU**2.0))
      DSTAR=CMPLX(DA,DA*(ETAA))
      MP=ROA*ACHA*LX*LY
C  THE SAMPLE SPACING IN WAVENUMBER SPACE IS HKX AND HKY
      HKX=KX1/N
      HKY=KY1/N
C   PARAMETERS FOR FFT2B INVERSE FOURIER TRANSFORM SUBROUTINE
      NRCOEF=N
      NCCOEF=N
      LDCOEF=N
      LDA=N
      WRITE(26,11) LX,LY
 11   FORMAT(' PARAMETERS OF FINITE PLATE',/,' DIMENSIONS OF PLATE'
     &,' (LX)  ',F8.3,' (LY) ',F8.3,'  METERS')
      WRITE(26,21) XO,YO
 21   FORMAT(' DRIVING POINT   XO ',F8.3,' YO ',F8.3,'
     & METERS')
      WRITE(26,31) F
 31   FORMAT(' DRIVING FORCE   ',F8.3,'  N')
      WRITE(26,41)ACHA
 41   FORMAT(' PLATE THICKNESS   ',F8.6,'  M')
      WRITE(26,51) ROA
 51   FORMAT(' PLATE DENSITY   ',F8.3,'  KG/CU-METERS')
      WRITE(26,61) EA
 61   FORMAT(' MODULUS  OF ELASTICITY ',E8.3,' N/SQ-METERS')
      WRITE(26,71) DA
 71   FORMAT(' BENDING RIGIDITY   ',E12.4)
      WRITE(26,81) ETAA
 81   FORMAT(' DAMPING COEFFICIENT',F8.4)
      WRITE(26,91)
 91   FORMAT(/,' PARAMETERS OF INFINITE PLATE',/)
      WRITE(26,101) ACHB
 101  FORMAT(' PLATE THICKNESS   ',F8.6,'  M')
      WRITE(26,111) ROB
 111  FORMAT(' PLATE DENSITY   ',F8.3,'  KG/CU-METERS')
      WRITE(26,121) EB
 121  FORMAT(' MODULUS OF ELASTICITY  ',E8.3,'  N/SQ-METERS')
      WRITE(26,131) DB
 131  FORMAT(' BENDING RIGIDITY   ',E12.4)
      WRITE(26,141) ETAB
 141  FORMAT(' DAMPING COEFICIENT',F8.4)
      WRITE(26,151) RO
 151  FORMAT(/,' PARAMETERS FOR FLUID',/,' DENSITY   ',F8.3,
     &'  KG/CU-METER')
      WRITE(26,161) MU
 161  FORMAT(' VISCOSITY   ',F8.7,'  KG/M-S')
      WRITE(26,181) C
 181  FORMAT(' SPEED OF SOUND   ',F8.3,'  M/S')
      WRITE(26,191) PRA
 191  FORMAT(' PRANDTL NUMBER   ',F8.3)
      WRITE(26,201) GAMMA
 201  FORMAT(' SPECIFIC HEAT RATIO   ',F8.3)
      WRITE(26,211) Z
 211  FORMAT(' DISTANCE BETWEEN PLATES   ',F8.3,'  M')
C INITIALIZATION ROUTINES FOR THE FFT
      CALL FFTCI(N,WFF1)
      CALL FFTCI(N,WFF2)
C    THE FREQUENCY RANGE IS TAKEN FROM OMEGAB TO OMEGAT IN INCREMENTS
C    CORRESPONDING TO INC
C
C     OMEGAB=10*IFIX(ALOG10(OMEGA1))
C     OMEGA2=INC*10*IFIX(ALOG10(OMEGA2))
C     DO 1000 OMEGAI=OMEGAB,OMEGAT
C     OMEGA=2.0*PI*(10.0**(OMEGAI/(INC*10.0)))
        DO 100 I=1,N
```

```
          DO 100 J=1,N
          AT(I,J)=0.0
 100      CONTINUE
C  K IS THE WAVENUMBER IN THE FLUID
          K=OMEGA/C
C  KC IS THE COMPLEX WAVENUMBER
          KC=(K)*CMPLX(1.0,(OMEGA*MU*((4.0/3.0)+((GAMMA-1.0)/PRA)
     &)/(2.0*RO*C**2.0)))
C  KP IS THE PLATE WAVENUNBER
          KP=(ROB*ACHB*OMEGA**2.0/DB)**.25
C  KPF IS THE COMPLEX WAVENUMBER**4.0
          KPF=(ROB*ACHB*OMEGA**2.0/DB)*CMPLX(1.0,ETAB)
          WRITE(26,12) OMEGA/(2.0*PI)
 12       FORMAT(/,' FREQUENCY  ',F8.3,'   HZ')
          WRITE(26,22) KC
 22       FORMAT(' COMPLEX WAVENUMBER  ',F8.3,' + i ',F8.3,' 1/M')
          WRITE(26,32) (KPF)**.25
 32       FORMAT(' COMPLEX PLATE WAVENUMBER  ',E12.4,' + i '
     &,E12.4,'  1/M')
C  THE NEXT SECTION (UNTIL LINE 58) COMPUTES THE UPPER AND LOWER
C  MODES WHICH THE EQUATION IS SUMMED OVER. NUP IS THE UPPER LIMIT
C  AND NLO IS THE LOWER LIMIT.
C  INITIALIZE AND INCREMENT NLO
          NLO=-1
 18       NLO=NLO+1
C  CALCULATE RESONANT OMEGA FOR THE EN=NLO, EM=NLO MODE
          OMEGAMN0=(DA/(ROA*ACHA))**.5*(((2.0*NLO+1.0)*PI/LX)**2.0
     &+((2.0*NLO+1.0)*PI/LY)**2.0)
C  IF THIS RESONANT OMEGA IS LARGER THAN OMEGA GO ON TO 28 IF NOT
C  INCREMENT NLO AND REPEAT
          IF (OMEGAMN0.GT.OMEGA) GOTO 28
          GOTO 18
C  INITIALIZE AND INCREMENT NUP
 28       NUP=-1
 38       NUP=NUP+1
C  CALCULATE THE RESONANT OMEGAS FOR EN=NUP, EM=0 AND EN=0, EM=NUP
C  MODES AND TEST IF LARGER THAN OMEGA
          OMEGAMN1=(DA/(ROA*ACHA))**.5*(((2.0*NUP+1.0)*PI/LX)**2.0
     &+(PI/LY)**2.0)
          OMEGAMN2=(DA/(ROA*ACHA))**.5*((PI/LX)**2.0
     &+((2.0*NUP+1.0)*PI/LY)**2.0)
C  IF LARGER THAN OMEGAGO ON TO 48, IF NOT INCREMENT NUP AND REPEAT
          IF((OMEGAMN1.GT.OMEGA).OR.(OMEGAMN2.GT.OMEGA)) GOTO 48
          GOTO 38
 48       CONTINUE
C  ADD THREE MODES ON TO UPPER LIMIT
          NUP=NUP+3
          N11=NLO
C  SUBTRACT THREE MODES FROM LOWER LIMIT, ZERO IS THE LOWEST
          NLO=NLO-3
          IF(N11.EQ.0) NLO=0
          IF(N11.EQ.1) NLO=0
          IF(N11.EQ.2) NLO=0
          WRITE (6,58) NLO,NUP
 58       FORMAT (2I5)
C  THE PLATE MODES ARE TAKEN FROM EM AND EN = NLO TO NUP
          DO 200 EM=NLO,NUP
          DO 200 EN=NLO,NUP
          DO 300 I=1,N
          DO 300 J=1,N
C  AT1 IS THE PREVIOUS MODE SUM
          AT1(I,J)=AT(I,J)
 300      CONTINUE
C  OMEGAMNSQ IS THE RESONANT OMEGA*2
          OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
```

```
        KXM=((2.0*EM+1.0)*PI/LX)
        KYN=((2.0*EN+1.0)*PI/LY)
C  COEFICIENT CALCULATES THE INPUT ARRAY (COEF) TO THE FFT
        CALL COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  F2T2B IS AN IMSL SUBROUTINE WHICH CALCULATES THE 2-D FFT OF
C  A SET OF FOURIER COEFFICIENTS.
C  COEF(I,J) IS THE INPUT ARRAY
C  A(I,J)     IS THE OUTPUT ARRAY
        CALL F2T2B(NRCOEF,NCCOEF,A,LDA,COEF,LDCOEF,WFF1,WFF2,
     &WC,CPY)
          DO 400 I=1,N
          DO 400 J=1,N
          A1(I,J)=(1/(2.0*PI)**2.0)*HKX*HKY*((-1)**I)*((-1)**J)
     &*A(I,J)*(COS((2.0*EM+1.0)*PI*XO/LX)*
     &COS((2.0*EN+1.0)*PI*YO/LY)/(OMEGAMNSQ-OMEGA**2.0))
          AT(I,J)=AT(I,J)+A1(I,J)
 400      CONTINUE
 200    CONTINUE
        WRITE (26,301)
 301    FORMAT (' PLATE RESPONSE')
          DO 500 I=1,N
          DO 500 J=1,N  8
          W(I,J)=((-4.0)*F*RO*OMEGA**2.0*AT(I,J))/(MP*DB)
          W1(I,J)=((-4.0)*F*RO*OMEGA**2.0*AT1(I,J))/(MP*DB)
          WREAL(I,J)=W(I,J)  8
          W1REAL(I,J)=W1(I,J)
C  WREAL(I,J) IS THE REAL PART OF THE PLATE RESPONSE
C  W(I,J)      IS THE COMPLEX PLATE RESPONSE
C  W1(I,J)     IS THE COMPLEX PLATE RESPONSE OF THE PREVIOUS MODE
 500      CONTINUE
C  THE FOLLOWING TWO DO-LOOPS CONVERT THE OUTPUT PLATE RESPONSE
C  IN THE X-Y PLANE SO THAT THE RESPONSE SHOWS THE CENTER OF THE
C  PLATE IN THE CENTER OF THE ARRAY
C
C                  1  2 | 1  2
C                  3  4 | 3  4
C              X  ------------
C                  1  2 | 1  2
C                  3  4 | 3  4
C                       Y
           DO 600 I=1,N
           DO 600 J=1,N/2
           TEMP1=WREAL(I,J)
           TEMP2=W1REAL(I,J)
           WREAL(I,J)=WREAL(I,N/2+J)
           W1REAL(I,J)=W1REAL(I,N/2+J)
           WREAL(I,N/2+J)=TEMP1
           W1REAL(I,N/2+J)=TEMP2
 600       CONTINUE
           DO 700 J=1,N
           DO 700 I=1,N/2
           TEMP1=WREAL(I,J)
           TEMP2=W1REAL(I,J)
           WREAL(I,J)=WREAL(N/2+I,J)
           W1REAL(I,J)=W1REAL(N/2+I,J)
           WREAL(N/2+I,J)=TEMP1
           W1REAL(N/2+I,J)=TEMP2
 700       CONTINUE
C  THE FOLLOWING DO LOOP ADDS ON THE FINAL POINT TO MAKE
C  THE MATRIX SYMETRICAL
           DO 550 I=1,N+1
           WREAL(I,N+1)=WREAL(I,1)
           WREAL(N+1,I)=WREAL(1,I)
 550       CONTINUE
C  OUTPUT PLATE RESPONSE
```

```
          DO 800 I=1,N
          DO 800 J=1,N
C         WRITE (26,401) I-1,J-1,W1REAL(I,J),I-1,J-1,WREAL(I,J)
C 401     FORMAT (' W1(',I4,',',I4,') = ',E12.4,2X,' W(',
C    &I4,',',I4,') = ',E12.4)
C  CALCULATE THE MAXIMUM, MINIMUM, AND INCREMENT FOR THE PLOT
          VMAX=AMAX1(ABS(WREAL(I,J)),ABS(VMAX))
          VMIN=-VMAX
          VINC=VMAX/2.0
 800      CONTINUE
C  PLOTTER CREATES A 3-D GRAPH OF THE PLATE RESPONSE
       CALL PLOTTER (VMAX,VMIN,VINC,WREAL,WORK1,WORK2,N,KX1,KY1,
    &Z,OMEGA,ACHB,ETAB,KP,K)
C 1000 CONTINUE
       END
C
       SUBROUTINE COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
    &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  COEFICIENT CALCULATES THE FOURIER COEFICIENTS FOR USE IN THE
C  FFT2B SUBROUTINE
       REAL KX(N),KY(N),IMN,HKX,HKY,PI,KX1,KY1,
    &KXM,KYN,Z,LX,LY,K
       INTEGER EM,EN,N
       COMPLEX TEMP,KC,KPF,COEF(N,N)
       DO 10 I=1,N
       DO 10 J=1,N
C  THE FFT USES THE INTERVAL FROM -KX1/2.0 TO KX1/2.0
       KX(I)=((I-1)*HKX)-(KX1/2.0)
       KY(J)=((J-1)*HKY)-(KY1/2.0)
C  THE FOLLOWING IF STATEMENTS DETERMINE THE VALUE FOR IMN
       IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
    &(KYN**2.0 .EQ. KY(J)**2.0))
    &IMN=LX*LY/4.0
       IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
    &(KYN**2.0 .NE. KY(J)**2.0))
    &IMN=LX*KYN*COS(KY(J)*LY/2.0)*(-1.0)**EN/
    &(KYN**2.0-KY(J)**2.0)
       IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
    &(KYN**2.0 .EQ. KY(J)**2.0))
    &IMN=LY*KXM*COS(KX(I)*LX/2.0)*(-1.0)**EM/
    &(KXM**2.0-KX(I)**2.0)
       IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
    &(KYN**2.0 .NE. KY(J)**2.0))
    &IMN=4.0*KXM*KYN*COS(KX(I)*LX/2.0)*
    &COS(KY(J)*LY/2.0)*(-1.0)**EM*(-1.0)**EN/
    &((KXM**2.0-KX(I)**2.0)*(KYN**2.0-
    &KY(J)**2.0))
C  THE FOLLOWING IF-STATEMENT TESTS IF THE EXPRESSION IS
C  NEGATIVE FOR USE IN THE COMPLEX EXPRESSION FOR COEF(I,J)
       IF (K**2.0-KX(I)**2.0-KY(J)**2.0) 55, 65, 65
 55    TEMP=CMPLX(((ABS(K**2.0-KX(I)**2.0-
    &KY(J)**2.0))**.5)*(-Z),0.0)
       GOTO 75
 65    TEMP=CMPLX(0.0,((K**2.0-KX(I)**2.0-
    &KY(J)**2.0)**0.5)*Z)
 75    CONTINUE
C  CALCULATE COEF
       COEF(I,J)=CEXP(TEMP)*IMN/
    &((KC**2.0-KX(I)**2.0-KY(J)**2.0)**.5*
    &((KX(I)**2.0+KY(J)**2.0)**2.0-KPF))
 10    CONTINUE
       RETURN
       END
C
       SUBROUTINE PLOTTER (VMAX,VMIN,VINC,WREAL,WORK1,WORK2,N,
    &KX1,KY1,Z,OMEGA,ACHB,ETAB,KP,K)
```

```
C  PLOTTER USES THE ARL_HIDE/TEMPLATE PLOTTING PACKAGE TO CREATE
C  3-D PLOTS OF THE PLATE RESPONSE
      REAL KX1,KY1,TXMIN,TYMIN,TXMAX,TYMAX,TXINC,TYINC
     &OMEGA,ACHB,ETAB,KP,K,Z,PI
      PI=2.0*ASIN(1.0)
      TXMIN=-N*PI/KX1
      TYMIN=-N*PI/KY1
      TXMAX=N*PI/KX1
      TYMAX=N*PI/KY1
      TXINC=N*PI/(4.0*KX1)
      TYINC=N*PI/(4.0*KY1)
      WRITE (6,501) KX1,TXMIN,TXMAX,TXINC,KY1,TYMIN,TYMAX,
     &TYINC,N
501   FORMAT (4F12.4,/,4F12.4,I4)
      CALL USLPDF
      CALL UPSET( 'OUTPUTFILE',7.0)
      CALL UASSGN(7.0, 'PR.PDF\')
      CALL USTART
      CALL UDIMEN( 9.0,6.5)
      CALL USET('PERCENT')
      CALL UFONT('CROM')
      CALL USET('LARGE')
      CALL UPRINT(61.,93.,'KX1= \')
      CALL UPRNT1(KX1,'REAL')
      CALL UPRINT(77.,93.,'KY1= \')
      CALL UPRNT1(KY1,'REAL')
      CALL UPRINT(61.,88.,'Z  = \')
      CALL UPRNT1(Z,'REAL')
      CALL UPRINT(77.,88.,'H  = \')
      CALL UPRNT1(ACHB,'REAL')
      CALL UPRINT(61.,83.,'FREQ=\')
      CALL UPRNT1(OMEGA/(2.0*PI),'REAL')
      CALL UPRINT(77.,83.,'OMEGA=\')
      CALL UPRNT1(OMEGA,'REAL')
      CALL UPRINT(61.,78.,'K  = \')
      CALL UPRNT1(K,'REAL')
      CALL UPRINT(77.,78.,'KP = \')
      CALL UPRNT1(KP,'REAL')
      CALL UPRINT(77.,73.,'ETA= \')
      CALL UPRNT1(ETAB,'REAL')
      CALL ARL_HIDE_FONT
      CALL UFONT('CROM')
      CALL USET('LARGE')
      CALL ARL_HIDE_SCALE(N+1,N+1,N+1,4000,N+1,VMIN,VMAX,
     &'CUTOFF',CUTOFF,0.5,0.6,1.25,0.5,0.5,0.0,45.0,45.0,
     &'FDRAW','SDRAW')
      CALL ARL_HIDE(WREAL,WORK1,WORK2)
      CALL ARL_HIDE_FAXIS('VIEW','REAL',TXMIN,TXMAX,TXINC,
     &'(F8.2)','X   METERS',1,50.0,50.0)
      CALL ARL_HIDE_SAXIS('VIEW','REAL',TYMIN,TYMAX,TYINC,
     &'(F8.2)','Y   METERS',1,50.0,50.0)
      CALL ARL_HIDE_VAXIS('VIEW','REAL',VMIN,VMAX,VINC,
     &'(E12.4)','AMPLITUDE',1,50.0,50.0)
      CALL UEND
      RETURN
      END
```

```
      PROGRAM PRTL5K
C  DETERMINES THE TRANSMISSION LOSS BETWEEN PLATE-A AND PLATE-B .
C  USING RMS VIBRATION LEVELS OF EACH PLATE
C
C  WRITTEN BY MICHAEL F. SHAW
C
      INTEGER OMEGAI, OMEGAB, OMEGAT, INC, EM, EN, I, J,
     &NRCOEF,NCCOEF,LDCOEF,LDA,N,NLO,NUP,N11
      PARAMETER (N=128)
      COMPLEX  DSTAR,OMEGAMNSQ,COEF(N,N),A(N,N),B1(N,N),
     &BT(N,N),KPF,KC,WA(N,N),WB(N,N),AT(N,N),B(N,N),
     &WC(N,N)
      REAL WFF1(4*N+15),WFF2(4*N+15),CPY(N,N)
      REAL WBREAL(N,N),WAREAL(N,N),WORK1(4000),WORK2(100),
     &AMPA,AMPB,TAU,TL(200),COUNT,OMEGA1,OMEGA2,X1(200),OMEGART
      REAL DA,DB,ETAA,ETAB,LX,LY,ROA,ROB,ACHA,ACHB,KX(N),KY(N),
     &KX1,KY1,KXM,KYN,F,RO,MP,XO,YO,IMN,Z,PRA,GAMMA,MU,NU,C,E,
     &HKX,HKY,PI,K,KP,OMEGA,OMEGAMN0,OMEGAMN1,OMEGAMN2
      INTRINSIC CMPLX
      EXTERNAL F2T2B,FFTCI
C THE NEXT TWO LINES ALLOW N TO BE LARGER THAN 64 POINTS
      COMMON /WORKSP/ RWKSP
      REAL RWKSP(8884)
C  ****************************************************************
C                      VARIABLE LIST
C
C   DA - BENDING RIGIDITY OF THE FINITE PLATE
C   DB -BENDING RIGIDITY OF THE INFINITE PLATE
C   ETAA - DAMPING COEFICIENT FOR THE FINITE PLATE
C   ETAB - DAMPING COEFICIENT FOR THE INFINITE PLATE
C   LX & LY - DIMENSIONS OF THE FINITE PLATE
C   ROA - DENSITY OF THE FINITE PLATE
C   ROB - DENSITY OF THE INFINITE PLATE
C   ACHA - THICKNESS OF THE FINITE PLATE
C   ACHB - THICKNESS OF THE INFINITE PLATE
C   RO - DENSITY OF FLUID
C   XO & YO - THE POINT WHERE THE INPUT FORCE F IS APPLIED
C   Z - DISTANCE BETWEEN THE PLATES
C   PRA - PRANDEL NUMBER FOR FLUID
C   GAMMA - SPECIFIC HEAT RATIO FOR FLUID
C   MU - VISCOSITY OF FLUID
C   NU - POISON'S RATIO
C   C - SPEED OF SOUND IN FLUID
C   EA - SHEER MODULUS OF THE FINITE PLATE
C   EB - SHEER MODULUS OF THE INFINITE PLATE
C   KX1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN X DIR
C   KY1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN Y DIR
C   ****************************************************************
      PARAMETER (F=1.0)
C    INPUT FORCE TO THE FINITE PLATE IS ASSUMED TO BE A UNIT FORCE (1)
      OPEN (UNIT=25, FILE='PR5K.DAT', STATUS='OLD')
      OPEN (UNIT=26, FILE='PRTL5K.OUT', STATUS='NEW')
C  ANOTHER LINE SO N CAN BE LARGER THAN 64
      CALL IWKIN(8884)
C  READ INPUT DATA FROM PR.DAT
      READ (25,15) LX,LY,XO,YO,ACHA,ACHB
 15   FORMAT(4F8.3,2F8.6)
      READ (25,25) ROA,ROB,RO,ETAA,ETAB,GAMMA,PRA
 25   FORMAT(2F8.2,F8.3,2F8.7,2F8.6)
      READ (25,35) MU,C,EA,EB,KX1,KY1
 35   FORMAT(F8.7,F8.3,2E8.3,2F8.2)
      READ (25,45) NU,OMEGA1,OMEGA2,INC,Z,OMEGA
 45   FORMAT(F8.4,F8.1,F16.1,I8,F8.3,F12.3)
C  COMPUTE THE VALUE OF PI
      PI=2.0*ASIN(1.0)
C  COMPUTE THE BENDING RIGIDITIES
```

```
      DA=(EA*ACHA**3.0)/(12.0*(1.0-NU**2.0))
      DB=(EB*ACHB**3.0)/(12.0*(1.0-NU**2.0))
      DSTAR=CMPLX(DA,DA*ETAA)
      MP=ROA*ACHA*LX*LY
      WRITE(26,11) LX,LY
 11   FORMAT(' PARAMETERS OF FINITE PLATE',/,' DIMENSIONS OF PLATE'
     &,' (LX)  ',F8.3,' (LY) ',F8.3,'  METERS')
      WRITE(26,21) XO,YO
 21   FORMAT(' DRIVING POINT   XO ',F8.3,' YO ',F8.3,'
     & METERS')
      WRITE(26,31) F
 31   FORMAT(' DRIVING FORCE  ',F8.3,'  N')
      WRITE(26,41)ACHA
 41   FORMAT(' PLATE THICKNESS   ',F8.6,'  M')
      WRITE(26,51) ROA
 51   FORMAT(' PLATE DENSITY   ',F8.3,'  KG/CU-METERS')
      WRITE(26,61) EA
 61   FORMAT(' MODULUS  OF ELASTICITY  ',E8.3,' N/SQ-METERS')
      WRITE(26,71) DA
 71   FORMAT(' BENDING RIGIDITY  ',E12.4)
      WRITE(26,81) ETAA
 81   FORMAT(' DAMPING COEFFICIENT',F8.4)
      WRITE(26,91)
 91   FORMAT(/,' PARAMETERS OF INFINITE PLATE',/)
      WRITE(26,101) ACHB
 101  FORMAT(' PLATE THICKNESS  ',F8.6,'  M')
      WRITE(26,111) ROB
 111  FORMAT(' PLATE DENSITY  ',F8.3,'  KG/CU-METERS')
      WRITE(26,121) EB
 121  FORMAT(' MODULUS OF ELASTICITY  ',E8.3,'  N/SQ-METERS')
      WRITE(26,131) DB
 131  FORMAT(' BENDING RIGIDITY  ',E12.4)
      WRITE(26,141) ETAB
 141  FORMAT(' DAMPING COEFICIENT',F8.4)
      WRITE(26,151) RO
 151  FORMAT(/,' PARAMETERS FOR FLUID',/,' DENSITY   ',F8.3,
     &'  KG/CU-METER')
      WRITE(26,161) MU
 161  FORMAT(' VISCOSITY  ',F8.7,'  KG/M-S')
      WRITE(26,181) C
 181  FORMAT(' SPEED OF SOUND   ',F8.3,'  M/S')
      WRITE(26,191) PRA
 191  FORMAT(' PRANDTL NUMBER  ',F8.3)
      WRITE(26,201) GAMMA
 201  FORMAT(' SPECIFIC HEAT RATIO  ',F8.3)
      WRITE(26,211) Z
 211  FORMAT(' DISTANCE BETWEEN PLATES  ',F8.3,'  M')
C
C    THE FREQUENCY RANGE IS TAKEN FROM 1258 TO 5000 Hz
      DO 1000 OMEGAI=31,37
      OMEGA=2.0*PI*(10.0**(OMEGAI/10.0))
C  KX1 AND KY1 ARE
C  THE LIMITS OF THE FFT. THE FFT IS TAKEN FROM -KX1/2.0 TO
C  KX1/2.0 AND FROM -KY1/2.0 TO KY1/2.0
      KX1=402.2
      KY1=402.2
C  THE SAMPLE SPACING IS HKX AND HKY
      HKX=KX1/N
      HKY=KY1/N
C   PARAMETERS FOR FFT2B INVERSE FOURIER TRANSFORM SUBROUTINE
      NRCOEF=N
      NCCOEF=N
      LDCOEF=N
      LDA=N
C  INITIALIZATION ROUTINES FOR THE FFT
      CALL FFTCI(N,WFF1)
```

```
      CALL FFTCI(N,WFF2)
C  K IS THE WAVENUMBER IN THE FLUID
      K=OMEGA/C
C  KC IS THE COMPLEX WAVENUMBER
      KC=(K)*CMPLX(1.0,(OMEGA*MU*((4.0/3.0)+((GAMMA-1.0)/PRA)
     &)/(2.0*RO*C**2.0)))
C  KP IS THE PLATE WAVENUMBER
      KP=(ROB*ACHB*OMEGA**2.0/DB)**.25
C  KPF IS THE COMPLEX PLATE WAVENUMBER**4.0
      KPF=(ROB*ACHB*OMEGA**2.0/DB)*CMPLX(1.0,ETAB)
      DO 100 I=1,N
      DO 100 J=1,N
      BT(I,J)=0.0
      AT(I,J)=0.0
100   CONTINUE
C  OMEGAMNSUB CALCULATES THE UPPER AND LOWER MODES NUP, NLO
      CALL OMEGAMNSUB(NLO,NUP,OMEGA,ROA,ACHA,DA,LX,LY)
C  THE PLATE MODES ARE TAKEN FROM EM AND EN = NLO TO NUP
      DO 200 EM=NLO,NUP
      DO 200 EN=NLO,NUP
C  OMEGANMSQ IS THE RESONANT OMEGA**2
      OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
      KXM=((2.0*EM+1.0)*PI/LX)
      KYN=((2.0*EN+1.0)*PI/LY)
C  COEFICIENT CALCULATES THE INPUT ARRAY TO THE FFT (COEF)
      CALL COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  FFT2B IS AN IMSL SUBROUTINE WHICH CALCULATES THE 2-D FFT OF
C  A SET OF FOURIER COEFFICIENTS
C  COEF(I,J) IS THE INPUT ARRAY
C  A(I,J)     IS THE OUTPUT ARRAY
      CALL F2T2B (NRCOEF,NCCOEF,A,LDA,COEF,LDCOEF,WFF1,WFF2,
     &WC,CPY)
        DO 400 I=1,N
        DO 400 J=1,N
        B(I,J)=A(I,J)
        B1(I,J)=(1.0/(2.0*PI)**2.0)*HKX*HKY*((-1)**I)*((-1)**J)
     &*B(I,J)*(COS((2.0*EM+1.0)*PI*XO/LX)*COS((2.0*EN+1.0)
     &*PI*YO/LY)/(OMEGAMNSQ-OMEGA**2.0))
        BT(I,J)=BT(I,J)+B1(I,J)
400     CONTINUE
200    CONTINUE
        DO 500 I=1,N
        DO 500 J=1,N  8
        WB(I,J)=((-1.0)*F*RO*OMEGA**2.0*BT(I,J))/(MP*DB)
        WBREAL(I,J)=WB(I,J)
C  WBREAL(I,J) IS THE REAL PART OF THE PLATE-B RESPONSE
C  WB(I,J)      IS THE COMPLEX PLATE-B RESPONSE
500     CONTINUE
C  THE FOLLOWING TWO DO-LOOPS CONVERT THE OUTPUT PLATE RESPONSE
C  IN THE X-Y PLANE SO THAT THE RESPONSE SHOWS THE CENTER OF THE
C  PLATE IN THE CENTER OF THE ARRAY
C
C                    1  2 | 1  2
C                    3  4 | 3  4
C               X   -------------
C                    1  2 | 1  2
C                    3  4 | 3  4
C                         Y
         DO 600 I=1,N
         DO 600 J=1,N/2
         TEMP=WBREAL(I,J)
         WBREAL(I,J)=WBREAL(I,N/2+J)
         WBREAL(I,N/2+J)=TEMP
600      CONTINUE
```

```
            DO 700 J=1,N
            DO 700 I=1,N/2
            TEMP=WBREAL(I,J)
            WBREAL(I,J)=WBREAL(N/2+I,J)
            WBREAL(N/2+I,J)=TEMP
 700        CONTINUE
         AMPB=0.0
         COUNT=0.0
C  CALCULATE THE AMPLITUDE OF VIBRATION IN THE PORTION OF THE PLATE
C  SHADOWED BY THE UPPER PLATE
           DO 800 I=(N/2)-(KX1*LX/(4.0*PI)),(N/2)+(KX1*LX/(4.0*PI))
           DO 800 J=(N/2)-(KY1*LY/(4.0*PI)),(N/2)+(KY1*LY/(4.0*PI))
         AMPB=AMPB+WBREAL(I,J)**2.0
         COUNT=COUNT+1.0
 800       CONTINUE
C  AMPB IS THE SUM OF THE SQUARES OF THE PLATE RESPONSE OF THE AREA
C  OF PLATE B SHADOWED BY PLATE A
C  AMPTB IS AMPB DIVIDED BY THE NUMBER OF POINTS
         AMPTB=AMPB/COUNT
C  CALCULATE THE VIBRATION RESPONSE OF PLATE A
         DO 250 EM=NLO,NUP
         DO 250 EN=NLO,NUP
         OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
           DO 450 I=1,N
           DO 450 J=1,N
           X=(-LX/2.0)+(LX*(I-1))/N
           Y=(-LY/2.0)+(LY*(J-1))/N
           A(I,J)=(COS((2.0*EM+1.0)*PI*XO/LX))*(COS((2.0*EN+1.0)
     &*PI*YO/LY))*(COS((2.0*EM+1.0)*PI*X/LX))*(COS((2.0*EN+1.0)
     &*PI*Y/LY))/(OMEGAMNSQ-OMEGA**2.0)
           AT(I,J)=AT(I,J)+A(I,J)
 450       CONTINUE
 250     CONTINUE
         AMPA=0.0
         COUNT=0.0
           DO 550 I=1,N
           DO 550 J=1,N
           WA(I,J)=(4.0*P/MP)*AT(I,J)
           WAREAL(I,J)=WA(I,J)
C  WAREAL IS THE REAL PART OF THE PLATE RESPONSE OF PLATE A
C  WA IS THE COMPLEX PLATE RESPONSE OF PLATE A
         AMPA=AMPA+WAREAL(I,J)**2.0
         COUNT=COUNT+1.0
 550     CONTINUE
         AMPTA=AMPA/COUNT
C  AMPA IS THE SUM OF THE SQUARES OF THE PLATE RESPONSE OVER PLATE A
C  AMPTA IS AMPA DIVIDED BY THE NUMBER OF POINTS
         WRITE (26,321) AMPTA,AMPTB
C  OUTPUT AMPTA AND AMPTB
 321     FORMAT (25X,' AMPTA=  ',E12.6,'  AMPTB=  ',E12.6)
C  THE FOLLOWING IF STATEMENT TESTS IF AMPTB IS TOO SMALL
         IF(AMPTB.LE.(10.0**(-90.0)))GO TO 650
         TAU=AMPTB/AMPTA
C  CALCULATE TRANSMISSION LOSS
         TL(OMEGAI)=10.0*(ALOG10(1.0/TAU))
         X1(OMEGAI)=FLOAT(OMEGAI)
C  OUTPUT TRANSMISSION LOSS
         WRITE(26,311)OMEGA/(2.0*PI),TL(OMEGAI)
 311     FORMAT(F12.2,10X,F6.2)
         GOTO 1000
 650     TL(OMEGAI)=101.0
         WRITE(26,311)OMEGA/(2.0*PI),TL(OMEGAI)
 1000    CONTINUE
         WRITE(26,411)
 411     FORMAT(//)
```

```
      END
C
      SUBROUTINE COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  COEFICIENT CALCULATES THE FOURIER COEFICIENTS FOR USE IN THE
C  FFT2B SUBROUTINE
      REAL KX(N),KY(N),IMN,HKX,HKY,PI,KX1,KY1,
     &KXM,KYN,Z,LX,LY,K
      INTEGER EM,EN,N
      COMPLEX TEMP,KC,KPF,COEF(N,N)
      DO 10 I=1,N
      DO 10 J=1,N
C  THE FFT USES THE INTERVAL FROM -KX1/2.0 TO KX1/2.0
C  AND -KY1/2.0 TO KY1/2.0
      KX(I)=((I-1)*HKX)-(KX1/2.0)
      KY(J)=((J-1)*HKY)-(KY1/2.0)
C  THE FOLLOWING IF STATEMENTS DETERMINE IMN
      IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
     &(KYN**2.0 .EQ. KY(J)**2.0))
     &IMN=LX*LY/4.0
      IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
     &(KYN**2.0 .NE. KY(J)**2.0))
     &IMN=LX*KYN*COS(KY(J)*LY/2.0)*(-1.0)**EN/
     &(KYN**2.0-KY(J)**2.0)
      IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
     &(KYN**2.0 .EQ. KY(J)**2.0))
     &IMN=LY*KXM*COS(KX(I)*LX/2.0)*(-1.0)**EM/
     &(KXM**2.0-KX(I)**2.0)
      IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
     &(KYN**2.0 .NE. KY(J)**2.0))
     &IMN=4.0*KXM*KYN*COS(KX(I)*LX/2.0)*
     &COS(KY(J)*LY/2.0)*(-1.0)**EM*(-1.0)**EN/
     &((KXM**2.0-KX(I)**2.0)*(KYN**2.0-
     &KY(J)**2.0))
C  THE FOLLOWING IF-STATEMENT TESTS IF THE EXPRESSION IS
C  NEGATIVE FOR USE IN THE COMPLEX EXXPRESSION FOR COEF(I,J)
      IF (K**2.0-KX(I)**2.0-KY(J)
     &**2.0) 55, 65, 65
 55   TEMP=CMPLX(((ABS(K**2.0-KX(I)**2.0-
     &KY(J)**2.0))**.5)*(-Z),0.0)
      GOTO 75
 65   TEMP=CMPLX(0.0,((K**2.0-KX(I)**2.0-
     &KY(J)**2.0)**0.5)*Z)
 75   CONTINUE
C CALCULATE COEF
      COEF(I,J)=CEXP(TEMP)*IMN/
     &((KC**2.0-KX(I)**2.0-KY(J)**2.0)**.5*
     &((KX(I)**2.0+KY(J)**2.0)**2.0-KPF))
 10   CONTINUE
      RETURN
      END
C
      SUBROUTINE OMEGAMNSUB(NLO,NUP,OMEGA,ROA,ACHA,DA,LX,LY)
C  OMEGAMNSUB COMPUTES THE UPPER AND LOWER MODES WHICH THE PLATE
C  RESPONSE IS SUMMED OVER. NLO IS THE LOWER LIMIT AND NUP IS THE
C  UPPER LIMIT.
      INTEGER NLO,NUP,N11
      REAL OMEGA,ROA,ACHA,DA,LX,LY,OMEGAMN0,OMEGAMN1,OMEGAMN2,PI
      PI=2.0*ASIN(1.0)
C  INITIALIZE AND INCREMENT NUP
      NLO=-1
 158  NLO=NLO+1
C  CALCULATE RESONANT OMEGA FOR THE EN=NLO, EM=NLO MODE
      OMEGAMN0=(DA/(ROA*ACHA))**.5*(((2.0*NLO+1.0)*PI/LX)**2.0
     &+((2.0*NLO+1.0)*PI/LY)**2.0)
C  IF THIS RESONANT OMEGA IS LARGER THAN OMEGA GO ON TO 258
```

```
C  IF NOT INCREMENT NLO AND REPEAT
         IF (OMEGAMN0.GT.OMEGA) GOTO 258
         GOTO 158
C  INITIALIZE AND INCREMENT NUP
 258     NUP=-1
 358     NUP=NUP+1
C  CALCULATE THE RESONANT OMEGAS FOR EN=NUP, EM=0 AND EN=0, EM=NUP
C  MODES AND TEST IF LARGER THAN OMEGA
         OMEGAMN1=(DA/(ROA*ACHA))**.5*(((2.0*NUP+1.0)*PI/LX)**2.0
     &+(PI/LY)**2.0)
         OMEGAMN2=(DA/(ROA*ACHA))**.5*((PI/LX)**2.0
     &+((2.0*NUP+1.0)*PI/LY)**2.0)
C  IF LARGER THAN OMEGA GO ON TO 458 IF NOT INCREMENT NUP , REPEAT
         IF((OMEGAMN1.GT.OMEGA).OR.(OMEGAMN2.GT.OMEGA)) GOTO 458
         GOTO 358
 458     CONTINUE
C  ADD THREE MODES TO UPPER LIMIT
         NUP=NUP+3
         N11=NLO
C  SUBTRACT THREE MODES FROM THE LOWER LIMIT, ZERO IS THE LOWEST NLO
         NLO=NLO-3
         IF(N11.EQ.0) NLO=0
         IF(N11.EQ.1) NLO=0
         IF(N11.EQ.2) NLO=0
         WRITE (6,558) NLO,NUP
 558     FORMAT (2I5)
         RETURN
         END
```

```
      PROGRAM PRTL1000
C  DETERMINES THE TRANSMISSION LOSS BETWEEN PLATE-A AND PLATE-B
C  USING RMS VIBRATION LEVELS OF EACH PLATE
C
C  WRITTEN BY MICHAEL F. SHAW
C
      INTEGER OMEGAI, OMEGAB, OMEGAT, INC, EM, EN, I, J,
     &NRCOEF,NCCOEF,LDCOEF,LDA,N,NLO,NUP,N11
      PARAMETER (N=128)
      COMPLEX  DSTAR,OMEGAMNSQ,COEF(N,N),A(N,N),B1(N,N),
     &BT(N,N),KPF,KC,WA(N,N),WB(N,N),AT(N,N),B(N,N),
     &WC(N,N)
      REAL WFF1(4*N+15),WFF2(4*N+15),CPY(N,N)
      REAL WBREAL(N,N),WAREAL(N,N),WORK1(4000),WORK2(100),
     &AMPA,AMPB,TAU,TL(200),COUNT,OMEGA1,OMEGA2,X1(200),OMEGART
      REAL DA,DB,ETAA,ETAB,LX,LY,ROA,ROB,ACHA,ACHB,KX(N),KY(N),
     &KX1,KY1,KXM,KYN,F,RO,MP,XO,YO,IMN,Z,PRA,GAMMA,MU,NU,C,E,
     &HKX,HKY,PI,K,KP,OMEGA,OMEGAMN0,OMEGAMN1,OMEGAMN2
      INTRINSIC CMPLX
      EXTERNAL F2T2B,FFTCI
C  THE NEXT TWO LINES ALLOW N TO BE LARGER THAN 64 POINTS
      COMMON /WORKSP/ RWKSP
      REAL RWKSP(8884)
C  ****************************************************************
C                       VARIABLE LIST
C
C   DA - BENDING RIGIDITY OF THE FINITE PLATE
C   DB -BENDING RIGIDITY OF THE INFINITE PLATE
C   ETAA - DAMPING COEFICIENT FOR THE FINITE PLATE
C   ETAB - DAMPING COEFICIENT FOR THE INFINITE PLATE
C   LX & LY - DIMENSIONS OF THE FINITE PLATE
C   ROA - DENSITY OF THE FINITE PLATE
C   ROB - DENSITY OF THE INFINITE PLATE
C   ACHA - THICKNESS OF THE FINITE PLATE
C   ACHB - THICKNESS OF THE INFINITE PLATE
C   RO - DENSITY OF FLUID
C   XO & YO - THE POINT WHERE THE INPUT FORCE F IS APPLIED
C   Z - DISTANCE BETWEEN THE PLATES
C   PRA - PRANDEL NUMBER FOR FLUID
C   GAMMA - SPECIFIC HEAT RATIO FOR FLUID
C   MU - VISCOSITY OF FLUID
C   NU - POISON'S RATIO
C   C - SPEED OF SOUND IN FLUID
C   EA - SHEER MODULUS OF THE FINITE PLATE
C   EB - SHEER MODULUS OF THE INFINITE PLATE
C   KX1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN X DIR
C   KY1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN Y DIR
C  ****************************************************************
      PARAMETER (F=1.0)
C   INPUT FORCE TO THE FINITE PLATE IS ASSUMED TO BE A UNIT FORCE (1)
      OPEN (UNIT=25, FILE='PR1000.DAT', STATUS='OLD')
      OPEN (UNIT=26, FILE='PRTL1000.OUT', STATUS='NEW')
C  ANOTHER LINE SO N CAN BE LARGER THAN 64
      CALL IWKIN(8884)
C  READ INPUT DATA FROM PR.DAT
      READ (25,15) LX,LY,XO,YO,ACHA,ACHB
 15   FORMAT(4F8.3,2F8.6)
      READ (25,25) ROA,ROB,RO,ETAA,ETAB,GAMMA,PRA
 25   FORMAT(2F8.2,F8.3,2F8.7,2F8.6)
      READ (25,35) MU,C,EA,EB,KX1,KY1
 35   FORMAT(F8.7,F8.3,2E8.3,2F8.2)
      READ (25,45) NU,OMEGA1,OMEGA2,INC,Z,OMEGA
 45   FORMAT(F8.4,F8.1,F16.1,I8,F8.3,F12.3)
C  COMPUTE THE VALUE OF PI
      PI=2.0*ASIN(1.0)
C  COMPUTE THE BENDING RIGIDITIES
```

```
      DA=(EA*ACHA**3.0)/(12.0*(1.0-NU**2.0))
      DB=(EB*ACHB**3.0)/(12.0*(1.0-NU**2.0))
      DSTAR=CMPLX(DA,DA*ETAA)
      MP=ROA*ACHA*LX*LY
      WRITE(26,11) LX,LY
11    FORMAT(' PARAMETERS OF FINITE PLATE',/,' DIMENSIONS OF PLATE'
     &,' (LX)  ',F8.3,' (LY) ',F8.3,'  METERS')
      WRITE(26,21) XO,YO
21    FORMAT(' DRIVING POINT   XO ',F8.3,' YO ',F8.3,'
     & METERS')
      WRITE(26,31) F
31    FORMAT(' DRIVING FORCE  ',F8.3,'  N')
      WRITE(26,41)ACHA
41    FORMAT(' PLATE THICKNESS  ',F8.6,'  M')
      WRITE(26,51) ROA
51    FORMAT(' PLATE DENSITY  ',F8.3,'  KG/CU-METERS')
      WRITE(26,61) EA
61    FORMAT(' MODULUS  OF ELASTICITY  ',E8.3,' N/SQ-METERS')
      WRITE(26,71) DA
71    FORMAT(' BENDING RIGIDITY   ',E12.4)
      WRITE(26,81) ETAA
81    FORMAT(' DAMPING COEFFICIENT',F8.4)
      WRITE(26,91)
91    FORMAT(/,' PARAMETERS OF INFINITE PLATE',/)
      WRITE(26,101) ACHB
101   FORMAT(' PLATE THICKNESS  ',F8.6,'  M')
      WRITE(26,111) ROB
111   FORMAT(' PLATE DENSITY  ',F8.3,'  KG/CU-METERS')
      WRITE(26,121) EB
121   FORMAT(' MODULUS OF ELASTICITY  ',E8.3,'  N/SQ-METERS')
      WRITE(26,131) DB
131   FORMAT(' BENDING RIGIDITY   ',E12.4)
      WRITE(26,141) ETAB
141   FORMAT(' DAMPING COEFICIENT',F8.4)
      WRITE(26,151) RO
151   FORMAT(/,' PARAMETERS FOR FLUID',/,' DENSITY  ',F8.3,
     &'  KG/CU-METER')
      WRITE(26,161) MU
161   FORMAT(' VISCOSITY  ',F8.7,'  KG/M-S')
      WRITE(26,181) C
181   FORMAT(' SPEED OF SOUND  ',F8.3,'  M/S')
      WRITE(26,191) PRA
191   FORMAT(' PRANDTL NUMBER  ',F8.3)
      WRITE(26,201) GAMMA
201   FORMAT(' SPECIFIC HEAT RATIO  ',F8.3)
      WRITE(26,211) Z
211   FORMAT(' DISTANCE BETWEEN PLATES  ',F8.3,'  M')
C  FREQUENCY IS TAKEN FROM 125 TO 1000 Hz
      DO 1000 OMEGAI=21,30
      OMEGA=2.0*PI*(10.0**(OMEGAI/10.0))
C  THE LIMITS OF THE FFT. THE FFT IS TAKEN FROM -KX1/2.0 TO
C  KX1/2.0 AND FROM -KY1/2.0 TO KY1/2.0
      KX1=201.1
      KY1=201.1
C  THE SAMPLE SPACING IS HKX AND HKY
      HKX=KX1/N
      HKY=KY1/N
C  PARAMETERS FOR FFT2B INVERSE FOURIER TRANSFORM SUBROUTINE
      NRCOEF=N
      NCCOEF=N
      LDCOEF=N
      LDA=N
C  INITIALIZATION ROUTINES FOR THE FFT
      CALL FFTCI(N,WFF1)
      CALL FFTCI(N,WFF2)
C  K IS THE WAVENUMBER IN THE FLUID
```

```
      K=OMEGA/C
C  KC IS THE COMPLEX WAVENUMBER
      KC=(K)*CMPLX(1.0,(OMEGA*MU*((4.0/3.0)+((GAMMA-1.0)/PRA)
     &)/(2.0*RO*C**2.0)))
C  KP IS THE PLATE WAVENUMBER
      KP=(ROB*ACHB*OMEGA**2.0/DB)**.25
C  KPF IS THE COMPLEX PLATE WAVENUMBER**4.0
      KPF=(ROB*ACHB*OMEGA**2.0/DB)*CMPLX(1.0,ETAB)
      DO 100 I=1,N
      DO 100 J=1,N
      BT(I,J)=0.0
      AT(I,J)=0.0
100   CONTINUE
C  OMEGAMNSUB CALCULATES THE UPPER AND LOWER MODES NUP, NLO
      CALL OMEGAMNSUB(NLO,NUP,OMEGA,ROA,ACHA,DA,LX,LY)
C  THE PLATE MODES ARE TAKEN FROM EM AND EN = NLO TO NUP
      DO 200 EM=NLO,NUP
      DO 200 EN=NLO,NUP
C  OMEGANMSQ IS THE RESONANT OMEGA**2
      OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
      KXM=((2.0*EM+1.0)*PI/LX)
      KYN=((2.0*EN+1.0)*PI/LY)
C  COEFICIENT CALCULATES THE INPUT ARRAY TO THE FFT (COEF)
      CALL COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  FFT2B IS AN IMSL SUBROUTINE WHICH CALCULATES THE 2-D FFT OF
C  A SET OF FOURIER COEFFICIENTS
C  COEF(I,J) IS THE INPUT ARRAY
C  A(I,J)     IS THE OUTPUT ARRAY
      CALL F2T2B (NRCOEF,NCCOEF,A,LDA,COEF,LDCOEF,WFF1,WFF2,
     &WC,CPY)
        DO 400 I=1,N
        DO 400 J=1,N
        B(I,J)=A(I,J)
        B1(I,J)=(1.0/(2.0*PI)**2.0)*HKX*HKY*((-1)**I)*((-1)**J)
     &*B(I,J)*(COS((2.0*EM+1.0)*PI*XO/LX)*COS((2.0*EN+1.0)
     &*PI*YO/LY)/(OMEGAMNSQ-OMEGA**2.0))
        BT(I,J)=BT(I,J)+B1(I,J)
400     CONTINUE
200   CONTINUE
        DO 500 I=1,N
        DO 500 J=1,N    8
        WB(I,J)=((-1.0)*F*RO*OMEGA**2.0*BT(I,J))/(MP*DB)
        WBREAL(I,J)=WB(I,J)
C  WBREAL(I,J) IS THE REAL PART OF THE PLATE-B RESPONSE
C  WB(I,J)      IS THE COMPLEX PLATE-B RESPONSE
500     CONTINUE
C  THE FOLLOWING TWO DO-LOOPS CONVERT THE OUTPUT PLATE RESPONSE
C  IN THE X-Y PLANE SO THAT THE RESPONSE SHOWS THE CENTER OF THE
C  PLATE IN THE CENTER OF THE ARRAY
C
C                  1  2 | 1  2
C                  3  4 | 3  4
C             X    -----------
C                  1  2 | 1  2
C                  3  4 | 3  4
C                       Y
         DO 600 I=1,N
         DO 600 J=1,N/2
         TEMP=WBREAL(I,J)
         WBREAL(I,J)=WBREAL(I,N/2+J)
         WBREAL(I,N/2+J)=TEMP
600      CONTINUE
         DO 700 J=1,N
         DO 700 I=1,N/2
```

```
          TEMP=WBREAL(I,J)
          WBREAL(I,J)=WBREAL(N/2+I,J)
          WBREAL(N/2+I,J)=TEMP
 700      CONTINUE
       AMPB=0.0
       COUNT=0.0
C  CALCULATE THE AMPLITUDE OF VIBRATION IN THE PORTION OF THE PLATE
C  SHADOWED BY THE UPPER PLATE
         DO 800 I=(N/2)-(KX1*LX/(4.0*PI)),(N/2)+(KX1*LX/(4.0*PI))
         DO 800 J=(N/2)-(KY1*LY/(4.0*PI)),(N/2)+(KY1*LY/(4.0*PI))
       AMPB=AMPB+WBREAL(I,J)**2.0
       COUNT=COUNT+1.0
 800     CONTINUE
C  AMPB IS THE SUM OF THE SQUARES OF THE PLATE RESPONSE OF THE AREA
C  OF PLATE B SHADOWED BY PLATE A
C  AMPTB IS AMPB DIVIDED BY THE NUMBER OF POINTS
       AMPTB=AMPB/COUNT
C  CALCULATE THE VIBRATION RESPONSE OF PLATE A
       DO 250 EM=NLO,NUP
       DO 250 EN=NLO,NUP
       OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
         DO 450 I=1,N
         DO 450 J=1,N
         X=(-LX/2.0)+(LX*(I-1))/N
         Y=(-LY/2.0)+(LY*(J-1))/N
         A(I,J)=(COS((2.0*EM+1.0)*PI*XO/LX))*(COS((2.0*EN+1.0)
     &*PI*YO/LY))*(COS((2.0*EM+1.0)*PI*X/LX))*(COS((2.0*EN+1.0)
     &*PI*Y/LY))/(OMEGAMNSQ-OMEGA**2.0)
         AT(I,J)=AT(I,J)+A(I,J)
 450     CONTINUE
 250   CONTINUE
       AMPA=0.0
       COUNT=0.0
         DO 550 I=1,N
         DO 550 J=1,N
         WA(I,J)=(4.0*F/MP)*AT(I,J)
         WAREAL(I,J)=WA(I,J)
C  WAREAL IS THE REAL PART OF THE PLATE RESPONSE OF PLATE A
C  WA IS THE COMPLEX PLATE RESPONSE OF PLATE A
       AMPA=AMPA+WAREAL(I,J)**2.0
       COUNT=COUNT+1.0
 550     CONTINUE
       AMPTA=AMPA/COUNT
C  AMPA IS THE SUM OF THE SQUARES OF THE PLATE RESPONSE OVER PLATE A
C  AMPTA IS AMPA DIVIDED BY THE NUMBER OF POINTS
       WRITE (26,321) AMPTA,AMPTB
C  OUTPUT AMPTA AND AMPTB
 321   FORMAT (25X,' AMPTA=  ',E12.6,'  AMPTB=  ',E12.6)
C  THE FOLLOWING IF STATEMENT TESTS IF AMPTB IS TOO SMALL
       IF(AMPTB.LE.(10.0**(-90.0)))GO TO 650
       TAU=AMPTB/AMPTA
C  CALCULATE TRANSMISSION LOSS
       TL(OMEGAI)=10.0*(ALOG10(1.0/TAU))
       X1(OMEGAI)=FLOAT(OMEGAI)
C  OUTPUT TRANSMISSION LOSS
       WRITE(26,311)OMEGA/(2.0*PI),TL(OMEGAI)
 311   FORMAT(F12.2,10X,F6.2)
       GOTO 1000
 650   TL(OMEGAI)=101.0
       WRITE(26,311)OMEGA/(2.0*PI),TL(OMEGAI)
 1000  CONTINUE
       END
C
       SUBROUTINE COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
```

```
C  COEFICIENT CALCULATES THE FOURIER COEFICIENTS FOR USE IN THE
C  FFT2B SUBROUTINE
       REAL KX(N),KY(N),IMN,HKX,HKY,PI,KX1,KY1,
      &KXM,KYN,Z,LX,LY,K
       INTEGER EM,EN,N
       COMPLEX TEMP,KC,KPF,COEF(N,N)
       DO 10 I=1,N
       DO 10 J=1,N
C  THE FFT USES THE INTERVAL FROM -KX1/2.0 TO KX1/2.0
C  AND -KY1/2.0 TO KY1/2.0
       KX(I)=((I-1)*HKX)-(KX1/2.0)
       KY(J)=((J-1)*HKY)-(KY1/2.0)
C  THE FOLLOWING IF STATEMENTS DETERMINE IMN
       IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
      &(KYN**2.0 .EQ. KY(J)**2.0))
      &IMN=LX*LY/4.0
       IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
      &(KYN**2.0 .NE. KY(J)**2.0))
      &IMN=LX*KYN*COS(KY(J)*LY/2.0)*(-1.0)**EN/
      &(KYN**2.0-KY(J)**2.0)
       IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
      &(KYN**2.0 .EQ. KY(J)**2.0))
      &IMN=LY*KXM*COS(KX(I)*LX/2.0)*(-1.0)**EM/
      &(KXM**2.0-KX(I)**2.0)
       IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
      &(KYN**2.0 .NE. KY(J)**2.0))
      &IMN=4.0*KXM*KYN*COS(KX(I)*LX/2.0)*
      &COS(KY(J)*LY/2.0)*(-1.0)**EM*(-1.0)**EN/
      &((KXM**2.0-KX(I)**2.0)*(KYN**2.0-
      &KY(J)**2.0))
C  THE FOLLOWING IF-STATEMENT TESTS IF THE EXPRESSION IS
C  NEGATIVE FOR USE IN THE COMPLEX EXXPRESSION FOR COEF(I,J)
       IF (K**2.0-KX(I)**2.0-KY(J)
      &**2.0) 55, 65, 65
 55    TEMP=CMPLX(((ABS(K**2.0-KX(I)**2.0-
      &KY(J)**2.0))**.5)*(-Z),0.0)
       GOTO 75
 65    TEMP=CMPLX(0.0,((K**2.0-KX(I)**2.0-
      &KY(J)**2.0)**0.5)*Z)
 75    CONTINUE
C CALCULATE COEF
       COEF(I,J)=CEXP(TEMP)*IMN/
      &((KC**2.0-KX(I)**2.0-KY(J)**2.0)**.5*
      &((KX(I)**2.0+KY(J)**2.0)**2.0-KPF))
 10    CONTINUE
       RETURN
       END
C
       SUBROUTINE OMEGAMNSUB(NLO,NUP,OMEGA,ROA,ACHA,DA,LX,LY)
C  OMEGAMNSUB COMPUTES THE UPPER AND LOWER MODES WHICH THE PLATE
C  RESPONSE IS SUMMED OVER. NLO IS THE LOWER LIMIT AND NUP IS THE
C  UPPER LIMIT.
       INTEGER NLO,NUP,N11
       REAL OMEGA,ROA,ACHA,DA,LX,LY,OMEGAMN0,OMEGAMN1,OMEGAMN2,PI
       PI=2.0*ASIN(1.0)
C  INITIALIZE AND INCREMENT NUP
       NLO=-1
 158   NLO=NLO+1
C  CALCULATE RESONANT OMEGA FOR THE EN=NLO, EM=NLO MODE
       OMEGAMN0=(DA/(ROA*ACHA))**.5*(((2.0*NLO+1.0)*PI/LX)**2.0
      &+((2.0*NLO+1.0)*PI/LY)**2.0)
C  IF THIS RESONANT OMEGA IS LARGER THAN OMEGA GO ON TO 258
C  IF NOT INCREMENT NLO AND REPEAT
       IF (OMEGAMN0.GT.OMEGA) GOTO 258
       GOTO 158
C  INITIALIZE AND INCREMENT NUP
```

```
 258    NUP=-1
 358    NUP=NUP+1
C  CALCULATE THE RESONANT OMEGAS FOR EN=NUP, EM=0 AND EN=0, EM=NUP
C  MODES AND TEST IF LARGER THAN OMEGA
        OMEGAMN1=(DA/(ROA*ACHA))**.5*(((2.0*NUP+1.0)*PI/LX)**2.0
       &+(PI/LY)**2.0)
        OMEGAMN2=(DA/(ROA*ACHA))**.5*((PI/LX)**2.0
       &+((2.0*NUP+1.0)*PI/LY)**2.0)
C  IF LARGER THAN OMEGA GO ON TO 458 IF NOT INCREMENT NUP , REPEAT
        IF((OMEGAMN1.GT.OMEGA).OR.(OMEGAMN2.GT.OMEGA)) GOTO 458
        GOTO 358
 458    CONTINUE
C  ADD THREE MODES TO UPPER LIMIT
        NUP=NUP+3
        N11=NLO
C  SUBTRACT THREE MODES FROM THE LOWER LIMIT, ZERO IS THE LOWEST NLO
        NLO=NLO-3
        IF(N11.EQ.0) NLO=0
        IF(N11.EQ.1) NLO=0
        IF(N11.EQ.2) NLO=0
        WRITE (6,558) NLO,NUP
 558    FORMAT (2I5)
        RETURN
        END
```

```
      PROGRAM PRTL100
C  DETERMINES THE TRANSMISSION LOSS BETWEEN PLATE-A AND PLATE-B
C  USING RMS VIBRATION LEVELS OF EACH PLATE
C
C  WRITTEN BY MICHAEL F. SHAW
C
      INTEGER OMEGAI, OMEGAB, OMEGAT, INC, EM, EN, I, J,
     &NRCOEF,NCCOEF,LDCOEF,LDA,N,NLO,NUP,N11
      PARAMETER (N=64)
      COMPLEX  DSTAR,OMEGAMNSQ,COEF(N,N),A(N,N),B1(N,N),
     &BT(N,N),KPF,KC,WA(N,N),WB(N,N),AT(N,N),B(N,N),
     &WC(N,N)
      REAL WFF1(4*N+15),WFF2(4*N+15),CPY(N,N)
      REAL WBREAL(N,N),WAREAL(N,N),WORK1(4000),WORK2(100),
     &AMPA,AMPB,TAU,TL(200),COUNT,OMEGA1,OMEGA2,X1(200),OMEGART
      REAL DA,DB,ETAA,ETAB,LX,LY,ROA,ROB,ACHA,ACHB,KX(N),KY(N),
     &KX1,KY1,KXM,KYN,F,RO,MP,XO,YO,IMN,Z,PRA,GAMMA,MU,NU,C,E,
     &HKX,HKY,PI,K,KP,OMEGA,OMEGAMN0,OMEGAMN1,OMEGAMN2
      INTRINSIC CMPLX
      EXTERNAL F2T2B,FFTCI
C  THE NEXT TWO LINES ALLOW N TO BE LARGER THAN 64 POINTS
      COMMON /WORKSP/ RWKSP
      REAL RWKSP(8884)
C  ****************************************************************
C                      VARIABLE LIST
C
C   DA - BENDING RIGIDITY OF THE FINITE PLATE
C   DB -BENDING RIGIDITY OF THE INFINITE PLATE
C   ETAA - DAMPING COEFICIENT FOR THE FINITE PLATE
C   ETAB - DAMPING COEFICIENT FOR THE INFINITE PLATE
C   LX & LY - DIMENSIONS OF THE FINITE PLATE
C   ROA - DENSITY OF THE FINITE PLATE
C   ROB - DENSITY OF THE INFINITE PLATE
C   ACHA - THICKNESS OF THE FINITE PLATE
C   ACHB - THICKNESS OF THE INFINITE PLATE
C   RO - DENSITY OF FLUID
C   XO & YO - THE POINT WHERE THE INPUT FORCE F IS APPLIED
C   Z - DISTANCE BETWEEN THE PLATES
C   PRA - PRANDEL NUMBER FOR FLUID
C   GAMMA - SPECIFIC HEAT RATIO FOR FLUID
C   MU - VISCOSITY OF FLUID
C   NU - POISON'S RATIO
C   C - SPEED OF SOUND IN FLUID
C   EA - SHEER MODULUS OF THE FINITE PLATE
C   EB - SHEER MODULUS OF THE INFINITE PLATE
C   KX1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN X DIR
C   KY1 - UPPER WAVENUMBER LIMIT IN FINITE PLATE IN Y DIR
C  ****************************************************************
      PARAMETER (F=1.0)
C   INPUT FORCE TO THE FINITE PLATE IS ASSUMED TO BE A UNIT FORCE (1)
      OPEN (UNIT=25, FILE='PR100.DAT', STATUS='OLD')
      OPEN (UNIT=26, FILE='PRTL100.OUT', STATUS='NEW')
C  ANOTHER LINE SO N CAN BE LARGER THAN 64
      CALL IWKIN(8884)
C  READ INPUT DATA FROM PR.DAT
      READ (25,15) LX,LY,XO,YO,ACHA,ACHB
 15   FORMAT(4F8.3,2F8.6)
      READ (25,25) ROA,ROB,RO,ETAA,ETAB,GAMMA,PRA
 25   FORMAT(2F8.2,F8.3,2F8.7,2F8.6)
      READ (25,35) MU,C,EA,EB,KX1,KY1
 35   FORMAT(F8.7,F8.3,2E8.3,2F8.2)
      READ (25,45) NU,OMEGA1,OMEGA2,INC,Z,OMEGA
 45   FORMAT(F8.4,F8.1,F16.1,I8,F8.3,F12.3)
C  COMPUTE THE VALUE OF PI
      PI=2.0*ASIN(1.0)
C  COMPUTE THE BENDING RIGIDITIES
```

```
      DA=(EA*ACHA**3.0)/(12.0*(1.0-NU**2.0))
      DB=(EB*ACHB**3.0)/(12.0*(1.0-NU**2.0))
      DSTAR=CMPLX(DA,DA*ETAA)
      MP=ROA*ACHA*LX*LY
      WRITE(26,11) LX,LY
11    FORMAT(' PARAMETERS OF FINITE PLATE',/,' DIMENSIONS OF PLATE'
     &,' (LX)  ',F8.3,' (LY) ',F8.3,'  METERS')
      WRITE(26,21) XO,YO
21    FORMAT(' DRIVING POINT   XO ',F8.3,' YO ',F8.3,'
     & METERS')
      WRITE(26,31) F
31    FORMAT(' DRIVING FORCE  ',F8.3,'  N')
      WRITE(26,41)ACHA
41    FORMAT(' PLATE THICKNESS  ',F8.6,'  M')
      WRITE(26,51) ROA
51    FORMAT(' PLATE DENSITY  ',F8.3,'  KG/CU-METERS')
      WRITE(26,61) EA
61    FORMAT(' MODULUS  OF ELASTICITY  ',E8.3,'  N/SQ-METERS')
      WRITE(26,71) DA
71    FORMAT(' BENDING RIGIDITY  ',E12.4)
      WRITE(26,81) ETAA
81    FORMAT(' DAMPING COEFFICIENT',F8.4)
      WRITE(26,91)
91    FORMAT(/,' PARAMETERS OF INFINITE PLATE',/)
      WRITE(26,101) ACHB
101   FORMAT(' PLATE THICKNESS  ',F8.6,'  M')
      WRITE(26,111) ROB
111   FORMAT(' PLATE DENSITY  ',F8.3,'  KG/CU-METERS')
      WRITE(26,121) EB
121   FORMAT(' MODULUS OF ELASTICITY  ',E8.3,'  N/SQ-METERS')
      WRITE(26,131) DB
131   FORMAT(' BENDING RIGIDITY  ',E12.4)
      WRITE(26,141) ETAB
141   FORMAT(' DAMPING COEFICIENT',F8.4)
      WRITE(26,151) RO
151   FORMAT(/,' PARAMETERS FOR FLUID',/,' DENSITY  ',F8.3,
     &'  KG/CU-METER')
      WRITE(26,161) MU
161   FORMAT(' VISCOSITY  ',F8.7,'  KG/M-S')
      WRITE(26,181) C
181   FORMAT(' SPEED OF SOUND  ',F8.3,'  M/S')
      WRITE(26,191) PRA
191   FORMAT(' PRANDTL NUMBER  ',F8.3)
      WRITE(26,201) GAMMA
201   FORMAT(' SPECIFIC HEAT RATIO  ',F8.3)
      WRITE(26,211) Z
211   FORMAT(' DISTANCE BETWEEN PLATES  ',F8.3,'  M')
C
C    THE FREQUENCY RANGE IS TAKEN FROM 10 TO 100 Hz
      WRITE(26,301)
301   FORMAT(' FREQUENCY (HZ)        TRANSMISSION LOSS ')
      DO 1000 OMEGAI=10,20
      OMEGA=2.0*PI*(10.0**(OMEGAI/10.0))
C  KX1 AND KY1  ARE
C  THE LIMITS OF THE FFT. THE FFT IS TAKEN FROM -KX1/2.0 TO
C  KX1/2.0 AND FROM -KY1/2.0 TO KY1/2.0
      KX1=62.9
      KY1=62.9
C  THE SAMPLE SPACING IS HKX AND HKY
      HKX=KX1/N
      HKY=KY1/N
C   PARAMETERS FOR FFT2B INVERSE FOURIER TRANSFORM SUBROUTINE
      NRCOEF=N
      NCCOEF=N
      LDCOEF=N
      LDA=N
```

```
C  INITIALIZATION ROUTINES FOR THE FFT
         CALL FFTCI(N,WFF1)
         CALL FFTCI(N,WFF2)
C  K IS THE WAVENUMBER IN THE FLUID
         K=OMEGA/C
C  KC IS THE COMPLEX WAVENUMBER
         KC=(K)*CMPLX(1.0,(OMEGA*MU*((4.0/3.0)+((GAMMA-1.0)/PRA)
     &)/(2.0*RO*C**2.0)))
C  KP IS THE PLATE WAVENUMBER
         KP=(ROB*ACHB*OMEGA**2.0/DB)**.25
C  KPF IS THE COMPLEX PLATE WAVENUMBER**4.0
         KPF=(ROB*ACHB*OMEGA**2.0/DB)*CMPLX(1.0,ETAB)
         DO 100 I=1,N
         DO 100 J=1,N
         BT(I,J)=0.0
         AT(I,J)=0.0
 100     CONTINUE
C  OMEGAMNSUB CALCULATES THE UPPER AND LOWER MODES NUP, NLO
         CALL OMEGAMNSUB(NLO,NUP,OMEGA,ROA,ACHA,DA,LX,LY)
C  THE PLATE MODES ARE TAKEN FROM EM AND EN = NLO TO NUP
         DO 200 EM=NLO,NUP
         DO 200 EN=NLO,NUP
C  OMEGANMSQ IS THE RESONANT OMEGA**2
         OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
         KXM=((2.0*EM+1.0)*PI/LX)
         KYN=((2.0*EN+1.0)*PI/LY)
C  COEFICIENT CALCULATES THE INPUT ARRAY TO THE FFT (COEF)
         CALL COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  FFT2B IS AN IMSL SUBROUTINE WHICH CALCULATES THE 2-D FFT OF
C  A SET OF FOURIER COEFFICIENTS
C  COEF(I,J) IS THE INPUT ARRAY
C  A(I,J)     IS THE OUTPUT ARRAY
         CALL F2T2B (NRCOEF,NCCOEF,A,LDA,COEF,LDCOEF,WFF1,WFF2,
     &WC,CPY)
          DO 400 I=1,N
          DO 400 J=1,N
          B(I,J)=A(I,J)
          B1(I,J)=(1.0/(2.0*PI)**2.0)*HKX*HKY*((-1)**I)*((-1)**J)
     &*B(I,J)*(COS((2.0*EM+1.0)*PI*XO/LX)*COS((2.0*EN+1.0)
     &*PI*YO/LY)/(OMEGAMNSQ-OMEGA**2.0))
          BT(I,J)=BT(I,J)+B1(I,J)
 400      CONTINUE
 200     CONTINUE
          DO 500 I=1,N
          DO 500 J=1,N    8
          WB(I,J)=((-1.0)*F*RO*OMEGA**2.0*BT(I,J))/(MP*DB)
          WBREAL(I,J)=WB(I,J)
C  WBREAL(I,J) IS THE REAL PART OF THE PLATE-B RESPONSE
C  WB(I,J)      IS THE COMPLEX PLATE-B RESPONSE
 500      CONTINUE
C  THE FOLLOWING TWO DO-LOOPS CONVERT THE OUTPUT PLATE RESPONSE
C  IN THE X-Y PLANE SO THAT THE RESPONSE SHOWS THE CENTER OF THE
C  PLATE IN THE CENTER OF THE ARRAY
C
C                    1  2 | 1  2
C                    3  4 | 3  4
C                X   -----------
C                    1  2 | 1  2
C                    3  4 | 3  4
C                         Y
          DO 600 I=1,N
          DO 600 J=1,N/2
          TEMP=WBREAL(I,J)
          WBREAL(I,J)=WBREAL(I,N/2+J)
```

```
          WBREAL(I,N/2+J)=TEMP
 600      CONTINUE
          DO 700 J=1,N
          DO 700 I=1,N/2
          TEMP=WBREAL(I,J)
          WBREAL(I,J)=WBREAL(N/2+I,J)
          WBREAL(N/2+I,J)=TEMP
 700      CONTINUE
       AMPB=0.0
       COUNT=0.0
C  CALCULATE THE AMPLITUDE OF VIBRATION IN THE PORTION OF THE PLATE
C  SHADOWED BY THE UPPER PLATE
         DO 800 I=(N/2)-(KX1*LX/(4.0*PI)),(N/2)+(KX1*LX/(4.0*PI))
         DO 800 J=(N/2)-(KY1*LY/(4.0*PI)),(N/2)+(KY1*LY/(4.0*PI))
       AMPB=AMPB+WBREAL(I,J)**2.0
       COUNT=COUNT+1.0
 800     CONTINUE
C  AMPB IS THE SUM OF THE SQUARES OF THE PLATE RESPONSE OF THE AREA
C  OF PLATE B SHADOWED BY PLATE A
C  AMPTB IS AMPB DIVIDED BY THE NUMBER OF POINTS
       AMPTB=AMPB/COUNT
C  CALCULATE THE VIBRATION RESPONSE OF PLATE A
       DO 250 EM=NLO,NUP
       DO 250 EN=NLO,NUP
       OMEGAMNSQ=DSTAR*((((2.0*EM+1.0)*PI/LX)**2.0+((2.0*EN+1.0)
     &*PI/LY)**2.0)**2.0)/(ROA*ACHA)
         DO 450 I=1,N
         DO 450 J=1,N
         X=(-LX/2.0)+(LX*(I-1))/N
         Y=(-LY/2.0)+(LY*(J-1))/N
         A(I,J)=(COS((2.0*EM+1.0)*PI*XO/LX))*(COS((2.0*EN+1.0)
     &*PI*YO/LY))*(COS((2.0*EM+1.0)*PI*X/LX))*(COS((2.0*EN+1.0)
     &*PI*Y/LY))/(OMEGAMNSQ-OMEGA**2.0)
         AT(I,J)=AT(I,J)+A(I,J)
 450     CONTINUE
 250   CONTINUE
       AMPA=0.0
       COUNT=0.0
         DO 550 I=1,N
         DO 550 J=1,N
         WA(I,J)=(4.0*F/MP)*AT(I,J)
         WAREAL(I,J)=WA(I,J)
C  WAREAL IS THE REAL PART OF THE PLATE RESPONSE OF PLATE A
C  WA IS THE COMPLEX PLATE RESPONSE OF PLATE A
       AMPA=AMPA+WAREAL(I,J)**2.0
       COUNT=COUNT+1.0
 550     CONTINUE
       AMPTA=AMPA/COUNT
C  AMPA IS THE SUM OF THE SQUARES OF THE PLATE RESPONSE OVER PLATE A
C  AMPTA IS AMPA DIVIDED BY THE NUMBER OF POINTS
       WRITE (26,321) AMPTA,AMPTB
C  OUTPUT AMPTA AND AMPTB
 321   FORMAT (25X,' AMPTA=  ',E12.6,'  AMPTB=  ',E12.6)
C  THE FOLLOWING IF STATEMENT TESTS IF AMPTB IS TOO SMALL
       IF(AMPTB.LE.(10.0**(-90.0)))GO TO 650
       TAU=AMPTB/AMPTA
C  CALCULATE TRANSMISSION LOSS
       TL(OMEGAI)=10.0*(ALOG10(1.0/TAU))
       X1(OMEGAI)=FLOAT(OMEGAI)
C  OUTPUT TRANSMISSION LOSS
       WRITE(26,311)OMEGA/(2.0*PI),TL(OMEGAI)
 311   FORMAT(F12.2,10X,F6.2)
       GOTO 1000
 650   TL(OMEGAI)=101.0
       WRITE(26,311)OMEGA/(2.0*PI),TL(OMEGAI)
 1000  CONTINUE
```

```
      END
C
      SUBROUTINE COEFICIENT (N,HKX,HKY,KX1,KY1,PI,KXM,KYN,
     &K,KC,KPF,Z,KX,KY,LX,LY,EM,EN,COEF)
C  COEFICIENT CALCULATES THE FOURIER COEFICIENTS FOR USE IN THE
C  FFT2B SUBROUTINE
      REAL KX(N),KY(N),IMN,HKX,HKY,PI,KX1,KY1,
     &KXM,KYN,Z,LX,LY,K
      INTEGER EM,EN,N
      COMPLEX TEMP,KC,KPF,COEF(N,N)
      DO 10 I=1,N
      DO 10 J=1,N
C  THE FFT USES THE INTERVAL FROM -KX1/2.0 TO KX1/2.0
C  AND -KY1/2.0 TO KY1/2.0
      KX(I)=((I-1)*HKX)-(KX1/2.0)
      KY(J)=((J-1)*HKY)-(KY1/2.0)
C  THE FOLLOWING IF STATEMENTS DETERMINE IMN
      IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
     &(KYN**2.0 .EQ. KY(J)**2.0))
     &IMN=LX*LY/4.0
      IF ((KXM**2.0 .EQ. KX(I)**2.0).AND.
     &(KYN**2.0 .NE. KY(J)**2.0))
     &IMN=LX*KYN*COS(KY(J)*LY/2.0)*(-1.0)**EN/
     &(KYN**2.0-KY(J)**2.0)
      IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
     &(KYN**2.0 .EQ. KY(J)**2.0))
     &IMN=LY*KXM*COS(KX(I)*LX/2.0)*(-1.0)**EM/
     &(KXM**2.0-KX(I)**2.0)
      IF ((KXM**2.0 .NE. KX(I)**2.0).AND.
     &(KYN**2.0 .NE. KY(J)**2.0))
     &IMN=4.0*KXM*KYN*COS(KX(I)*LX/2.0)*
     &COS(KY(J)*LY/2.0)*(-1.0)**EM*(-1.0)**EN/
     &((KXM**2.0-KX(I)**2.0)*(KYN**2.0-
     &KY(J)**2.0))
C  THE FOLLOWING IF-STATEMENT TESTS IF THE EXPRESSION IS
C  NEGATIVE FOR USE IN THE COMPLEX EXXPRESSION FOR COEF(I,J)
      IF (K**2.0-KX(I)**2.0-KY(J)
     &**2.0) 55, 65, 65
 55   TEMP=CMPLX(((ABS(K**2.0-KX(I)**2.0-
     &KY(J)**2.0))**.5)*(-Z),0.0)
      GOTO 75
 65   TEMP=CMPLX(0.0,((K**2.0-KX(I)**2.0-
     &KY(J)**2.0)**0.5)*Z)
 75   CONTINUE
C CALCULATE COEF
      COEF(I,J)=CEXP(TEMP)*IMN/
     &((KC**2.0-KX(I)**2.0-KY(J)**2.0)**.5*
     &((KX(I)**2.0+KY(J)**2.0)**2.0-KPF))
 10   CONTINUE
      RETURN
      END
C
      SUBROUTINE OMEGAMNSUB(NLO,NUP,OMEGA,ROA,ACHA,DA,LX,LY)
C  OMEGAMNSUB COMPUTES THE UPPER AND LOWER MODES WHICH THE PLATE
C  RESPONSE IS SUMMED OVER. NLO IS THE LOWER LIMIT AND NUP IS THE
C  UPPER LIMIT.
      INTEGER NLO,NUP,N11
      REAL OMEGA,ROA,ACHA,DA,LX,LY,OMEGAMN0,OMEGAMN1,OMEGAMN2,PI
      PI=2.0*ASIN(1.0)
C  INITIALIZE AND INCREMENT NUP
      NLO=-1
 158  NLO=NLO+1
C  CALCULATE RESONANT OMEGA FOR THE EN=NLO, EM=NLO MODE
      OMEGAMN0=(DA/(ROA*ACHA))**.5*(((2.0*NLO+1.0)*PI/LX)**2.0
     &+((2.0*NLO+1.0)*PI/LY)**2.0)
C  IF THIS RESONANT OMEGA IS LARGER THAN OMEGA GO ON TO 258
```

```
C  IF NOT INCREMENT NLO AND REPEAT
         IF (OMEGAMN0.GT.OMEGA) GOTO 258
         GOTO 158
C  INITIALIZE AND INCREMENT NUP
 258     NUP=-1
 358     NUP=NUP+1
C  CALCULATE THE RESONANT OMEGAS FOR EN=NUP, EM=0 AND EN=0, EM=NUP
C  MODES AND TEST IF LARGER THAN OMEGA
         OMEGAMN1=(DA/(ROA*ACHA))**.5*(((2.0*NUP+1.0)*PI/LX)**2.0
     &+(PI/LY)**2.0)
         OMEGAMN2=(DA/(ROA*ACHA))**.5*((PI/LX)**2.0
     &+((2.0*NUP+1.0)*PI/LY)**2.0)
C  IF LARGER THAN OMEGA GO ON TO 458 IF NOT INCREMENT NUP , REPEAT
         IF((OMEGAMN1.GT.OMEGA).OR.(OMEGAMN2.GT.OMEGA)) GOTO 458
         GOTO 358
 458     CONTINUE
C  ADD THREE MODES TO UPPER LIMIT
         NUP=NUP+3
         N11=NLO
C  SUBTRACT THREE MODES FROM THE LOWER LIMIT, ZERO IS THE LOWEST NLO
         NLO=NLO-3
         IF(N11.EQ.0) NLO=0
         IF(N11.EQ.1) NLO=0
         IF(N11.EQ.2) NLO=0
         WRITE (6,558) NLO,NUP
 558     FORMAT (2I5)
         RETURN
         END
```

Sample Input

```
   1.000   1.000   0.000  0.0000.0015870.001587
7860.0007860.000   1.204 0.10000 0.10000  1.4000    .706
.0000184 343.000 200.0E9 200.0E9   402.1   402.1
  0.2900    10.0         10000.0       1   0.100    21020.0
```